HEAT TRANSFER AND EVAPORATION

BY

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To My Father

WHOSE PATIENT ANSWERS TO A SMALL
BOY'S QUESTIONS ROUSED THE
INTERESTS THAT HAVE RESULTED IN THIS BOOK

Preface.

If I may have a personal and informal word with my readers before the serious business of this work begins, I wish to assure them that I am fully sensible of all the shortcomings of this book.

It was begun with the ambitious intention of a comprehensive work on heat transfer in all its phases. This intention lasted through Chapter 4, weakened during Chapter 5, and disappeared entirely after Chapter 6. These six chapters had taken three years, and it was obvious that something must be done to quicken the pace. From this point on, the book is devoted strictly to evaporators. The result of this change in plan is a lack of continuity and a decided change in method of presentation.

With the change in plan, there came also a change in the type of audience I wished to interest. The first chapters were aimed at the student and research worker, the later chapters are for designing and operating engineers. If the latter class find the first few chapters aimless and stupid from their point of view, they may begin with Chapter 7 and lose little, if they are willing to take a very few fundamental equations on faith.

The book is not intended as a manual for unskilled operators. No excuse is needed or offered for the use of mathematics—in fact, I apologize for not having given a mathematical treatment of more of the subjects, and especially for not having worked more problems. Since the men who design evaporators are few in number and, in many respects, better qualified than I along these lines, they have been less in my mind as I have written than those who plan and operate processes in which evaporators are used. From such men I ask suggestions as to how the book might be made more useful to them.

Patent references have been made fairly full, back to 1905. Between then and the present the references are perhaps 75 per cent complete; though in discussing evaporator types, patents on obsolete machines or machines that never existed except on paper have not been cited.

The principal contributions I have to offer from my own experience and research are in Chapters 9 and 10. I cannot state too strongly that nothing will so surely lead to a thorough understanding of what goes on in a multiple effect, as the first-hand solution of such problems as worked in Chapter 10. The fact that data are only available for one type of evaporator and one set of operating conditions has no bearing on the fact that the general qualitative results of such calculations are most illuminating.

For assistance in the preparation of this book I should thank many persons and companies. The Swenson Evaporator Company has been mainly responsible for the book; in that they have given so freely of their information, and permitted me to spend in its preparation so much of time which belonged to them. They have also permitted the use of much unpublished research work. In particular, I should mention the assistance of Mr. P. B. Sadtler and Mr. H. D. Grant, of their staff. Where manufacturers of equipment have lent cuts, I have made acknowledgment in the text. For the bulk of the illustrations, however, credit must be given to my brother, Mr. P. A. Badger. Finally, the book could never have been written without the help of a succession of enthusiastic students, who carried out the experimental work quoted, helped develop the theory, checked calculations; in short, without whom the original contributions I have to make to this subject could never have been developed.

W. L. BADGER.

Ann Arbor, Michigan,
July 27, 1925.

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Chapter I.

General Theory of Heat Transmission—Conduction.

Heat may flow from a warmer to a cooler body by any of three methods: conduction, convection, or radiation. In most engineering processes the first two methods, and in many processes all three, occur simultaneously.

They may be distinguished as follows:

Conduction. When heat flows from a warmer to a cooler part of a body without physical transfer of material, the flow is said to be by conduction. So long as the medium is uniform in physical properties, heat transfer per unit distance is proportional to the temperature difference.

Convection. When heat flows from a warmer to a cooler part of a fluid by physical transfer of material, the process is called convection.

Radiation. When heat flows from a warmer to a cooler body by a process which would go on in essentially the same manner were there no medium between, the process is called radiation. The flow of heat through any medium or through a vacuum by this method is proportional to the 4th power of the temperature drop.

Thermal Conductivity.

In the discussion of conductivity, the following statements are assumed as axiomatic (for uniform flow in a conductor of uniform physical properties):

(*a*) The quantity of heat flowing is directly proportional to the cross-section of the conductor.

(*b*) The quantity of heat flowing is inversely proportional to the length of the path of flow.

(*c*) The quantity of heat flowing is directly proportional to the difference in temperature across the section of the path of flow under consideration. These may be expressed mathematically as

$$Q = \frac{\lambda a (t_2 - t_1)}{L} \qquad (1)$$

where Q is the total quantity of heat, a the area of the path, t_2 and t_1 the temperatures of the hot and cold ends of the path respectively, L the length of the path, and λ a factor which represents a specific property of the material in question.

If we make a, $(t_2 - t_1)$, and L, all equal to unity, the quantity of heat which would then flow is equal to the numerical value of λ and is called the *thermal conductivity* of the material. In scientific work, this is usually expressed as calories per square centimeter of area per centimeter of length per degree C. per second.[1] In English units it is usually expressed as B.t.u. per square foot of area per foot of length per degree F. per hour. To convert data in the C.G.S. system to English units multiply by 242.

Conductivity of Solids.

The principal difficulty in determining the conductivity of any material is that conductivity changes with composition. In few cases have experimenters sufficiently described their materials, and in no case is the relation between conductivity and composition known. For instance, two samples of 0.1 per cent carbon steel, each examined by a different careful experimenter, differ by nearly 10 per cent: yet "0.1 per cent C steel" is a much more accurate definition of the material than "cast iron." Hence, in the following table, it is to be understood that differences in composition within ordinary commercial limits may cause a variation of from 10 to 25 per cent or even more in thermal conductivity.[2]

It will be seen that thermal conductivity of metals changes little with temperature in the ordinary range. In most cases there is a greater change at low temperatures, and less change above room temperature.

As will be shown later, the conductivity of metal plays so small a part in the ordinary cases of heat transmission from one fluid to another that the above values are amply accurate.

Conductivity of Liquids.

The conductivity of liquids is best handled by an expression developed by Weber[3] which permits the calculation of thermal conductivity from constants which are more easily derived. It is

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[1] Data are sometimes given as watts per second instead of calories per second. To convert to calories, multiply watts by 0.239. In engineering work in metric units, conductivity is usually given as large calories per square meter per meter per ° C. per hour. To convert this to the scientific unit, divide by 360; to convert to English units, multiply by 0.673.

[2] Figures in the table marked with an asterisk are obtained by a comparison of the work of Jaeger and Diesselhorst, *Wiss. Abh. Phys.-Tech. Reichsanstalt, 3,* 269-424 (1900); Lees, *Phil. Trans., 208* (*A*), 381-443 (1908); and Meissner, *Ann. Phys., 47,* 1001-58 (1915). Lees and Meissner worked from the temperature of liquid air to room temperature, Jaeger and Diesselhorst from room temperature to 212°. They agree fairly well, especially on such metals as copper, lead, silver, tin, and zinc, which are usually quite pure. Their values for iron and steel differ somewhat more, and their values for brass or bronze are widely different, due to differences in composition of the materials.

Other figures in the table are taken from Landolt Börnstein, "Phys Chem. Tabellen," or from trade bulletins, and have not been studied further. They are to be considered as rough approximations only.

[3] *Wied. Ann., 48,* 173 (1893). See McAdams and Frost, *J. Ind. Eng. Chem., 14,* 17 (1922).

$$\lambda = 0.868\ CS \sqrt[3]{\frac{S}{M}} \tag{2}$$

where C is the specific heat of the liquid, S is its specific gravity, and M its molecular weight. For water this gives 0.332 at room temperature in English units, which compares very well with the best experimental values. Jakob[4] gives, in C.G.S. units, $\lambda = 0.001325\ (1 + 0.002984t)$ and states that this is probably accurate to 1 per cent. This has been calculated in English units in Table II.

Conductivity of Gases.

This is a much more uncertain quantity than those previously discussed. From the kinetic theory of gases, it may be shown that the conductivity of a gas should be independent of the pressure down to pressures at which the mean free path of the gas molecules is comparable with the thickness of the gas layer through which heat is flowing. This is reached at a pressure of about 10 centimeters absolute (4").[5]

TABLE I

THERMAL CONDUCTIVITY OF SOLIDS

(In B.t.u. per square foot per foot per ° F. per hour)

	32° F.	70° F.	212° F
* Aluminum	118.0	118.0	121.0
* Brass (70-30)	60.5	63.0	72.5
Brass (unspecified)	....	48 to 63	....
* Bronze (Cu 85.7 per cent, Zn 7.15 per cent, Sn 6.4 per cent)	31.5	34.0	41.0
* Copper	230.0	227.0	220.0
Duriron	....	30.0	....
* Iron (wrought)	35.0	35.0	35.0
Iron (cast)	....	22 to 36	....
* Lead	20.1	19.8	19.6
Monel	....	15.0	....
* Nickel	34.0	34.0	34.0
* Silver	237.0	239.0	240.0
* Steel (0.1 per cent C)	26.2	26.1	25.8
* Tin	38.5	38.5	34.0
* Zinc	65.5	65.5	63.0
Calcium Carbonate (boiler scale)	....	0.5 to 1.5	....
Calcium Sulfate	0.4 to 0.75	....	....
Glass	....	0.26 to 0.55	....
Hard Rubber	....	0.021	....
Petroleum and Lubricating Oils	....	0.7 to 0.85	....
Porcelain	....	0.6	....
Sodium Chloride	0.33	....	....

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[4] *Ann. Phys., 63* (6), 537-70 (1920).

[5] But see Wamsler (no change between 1.5 and 0.095 mm.), *Mitt. Forsch.*, No. 98, 15; and Kundt and Warburg, *Pogg. Ann., 156,* 177 (1875) (not reached at 0.5 mm.).

TABLE II

Thermal Conductivity of Water

(English Units)

Temperature, ° F.	Conductivity	Temperature, ° F.	Conductivity
32	0.321	130	0.373
35	0.322	135	0.376
40	0.325	140	0.378
45	0.328	145	0.381
50	0.330	150	0.383
55	0.333	155	0.386
60	0.336	160	0.389
65	0.338	165	0.391
70	0.341	170	0.394
75	0.343	175	0.397
80	0.346	180	0.399
85	0.349	185	0.402
90	0.351	190	0.405
95	0.354	195	0.407
100	0.357	200	0.410
105	0.359	205	0.413
110	0.362	210	0.416
115	0.365	212	0.417
120	0.368	215	0.419
125	0.371		

The data in Table III are obtained by a comparison of a number of results,[6] and are probably correct to better than 5 per cent.

TABLE III

Thermal Conductivity of Gases

(B.t.u. per square foot per foot per ° F. per hour)

	32° F.	212° F.	Increase per ° F.
Air	0.0138	0.0174	0.000020
Ammonia	0.0111	0.0172	0.000034
Carbon dioxide	0.0082	0.0119	0.000020
Hydrogen	0.0970	0.1230	0.000139
Methane	0.0176	....	....
Nitrogen	0.0137	0.0173	0.000020
Oxygen	0.0138	0.0174	0.000020

Such data are of little value, for in most cases of heat transfer through gases, the prinicipal phenomena are radiation and convection.

Conductivities in Series and Parallel.

Conductivity and its reciprocal, resistivity, obey the same laws as the corresponding electrical units. In electricity, however, we are more accustomed to resistance and its reciprocal, conductance. Conductance is related to conductivity by the expression $K = \frac{\lambda}{L}$, where L is the length of conductor considered. Resistance is then $\frac{1}{K}$ and resistivity $\frac{1}{\lambda}$.

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[6] These data, with the exception of ammonia, were obtained by a comparison of the work of Schliermacher, *Wied. Ann.*, *34*, 623 (1888); Winkelmann, *Wied. Ann.*, *48*, 80 (1893); Schwarze, *Ann. Phys.*, *11* (4), 303 (1893); Muller, *Wied. Ann.*, *60*, 82 (1897); Eckerlein, *Ann. Phys.*, *3* (4), 120 (1900); Compau, *Compt. Rend.*, *133*, 1202 (1901); Todd, *Proc. Royal Soc.*, *83*(*A*), 19-39 (1909); Eucken, *Phys. Ztschr.*, *12*, 1101-7 (1911); and Weber, *Ann. Phys.*, *54* (4), 326, 437 (1917).

By analogy from the laws for electrical circuits, we have, where several resistances are in series, $R = r_1 + r_2 + r_3$. But since $r_1 = \frac{L_1}{\lambda_1}$, etc. (where r_1, r_2, etc., are the resistances of the several parts, and L_1, L_2, etc., are the corresponding distances), we have

$$\frac{1}{R} = \frac{1}{\frac{L_1}{\lambda_1} + \frac{L_2}{\lambda_2} + \frac{L_3}{\lambda_3}} \tag{3}$$

$\frac{1}{R}$, the overall conductance, is usually called the "overall coefficient" and is represented by U.

Where resistances are in parallel, the conductances may be added, and $U = K_1 + K_2 + K_3$, from which it follows that

$$\frac{\lambda}{L} = \frac{\lambda_1}{L_1} + \frac{\lambda_2}{L_2} + \frac{\lambda_3}{L_3} \tag{4}$$

Problem.

Consider a case of resistances in series. A plate of copper is 0.1 inch thick and has a layer of water 0.01 inch thick on one side and a layer of air 0.01 inch thick on the other. These films of air and water are transmitting heat by pure conduction. We shall see later that cases similar to this are very important in practice. Let us further suppose that the whole system is at about 212° F. with a temperature drop of 1° F. across the three layers. Then, considering one square foot, we have

for water; $\lambda_1 = 0.418$, $L_1 = 0.01$ inch $= 0.00083$ foot
for copper; $\lambda_2 = 220$, $L_2 = 0.10$ inch $= 0.0083$ foot
for air; $\lambda_3 = 0.0174$, $L_3 = 0.01$ inch $= 0.00083$ foot

Substituting in equation (3) and solving gives $U = 20.1$.

Temperature Drops across Compound Resistances.

In the above problem, it might be of interest to calculate the temperature drops across each of the films. This is done by considering the significance of equation (1) for compound resistances. If we limit the discussion to one square foot of surface, and let t_1, t_2, t_3, and t_4 be the temperature of the outside surface of the water, the contact surface between water and copper, the contact between copper-air, and outside surface of air film, respectively, we have

$$Q = \frac{\lambda_1 (t_1 - t_2)}{L_1} = \frac{\lambda_2 (t_2 - t_3)}{L_2} = \frac{\lambda_3 (t_3 - t_4)}{L_3}$$

Let us write Θ_1, Θ_2, Θ_3 for $(t_1 - t_2)$, $(t_2 - t_3)$, and $(t_3 - t_4)$.

Then

$$\Theta_1 = Q\frac{L_1}{\lambda_1},\ \Theta_2 = Q\frac{L_2}{\lambda_2},\ \text{and}\ \Theta_3 = Q\frac{L_3}{\lambda_3} \tag{5}$$

But $\frac{L_1}{\lambda_1} = r_1$, $\frac{L_2}{\lambda_2} = r_2$, etc.; hence $\Theta_1 = Qr_1$, $\Theta_2 = Qr_2$, and $\Theta_1 : \Theta_2 : \Theta_3 :: r_1 : r_2 : r_3 :: \frac{1}{K_1} : \frac{1}{K_2} : \frac{1}{K_3}$, etc.

Or in other words, in strict analogy to electrical circuits, the fall of temperature along a conductor is proportional to resistance. By substituting the proper values from the problem on page 13, we find $\Theta_1 = 0.0402°$, $\Theta_2 = 0.00076°$, and $\Theta_3 = 0.959°$.

Heat Transfer through Tube Walls.

In most cases in practice, the thickness of a tube wall is considered to be small enough in proportion to the diameter so that the curvature of the surface need not be considered. In case this is not true, the flow of heat may be determined by the following method: Let Figure 1 be a section of a tube of inside radius r_1 and outside radius r_2. Let the temperature of the outside of the metal be t_2 and of the inside be t_1. Let t_2 be greater than t_1, so that heat flows to the inside. Assume a piece of tube 1 foot long. Let us consider the flow of heat through a section whose thickness is dr and which is at a distance r from the center. The temperature drop across this ring will be dt. The sur-

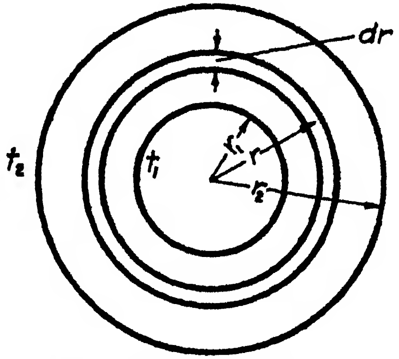

FIG. 1.—Flow of heat through thick-walled tube.

face of the ring will be $2\pi r$ square feet. Substituting in equation (1) we have

$$Q = \frac{2\pi r\lambda dt}{dr}$$

from which $dt = \frac{Qdr}{2\pi r\lambda}$ and $t_2 - t_1 = \int_{r_1}^{r_2} \frac{Qdr}{2\pi r\lambda} = \frac{Q}{2\pi\lambda}\int_{r_1}^{r_2} \frac{dr}{r}$

$$\text{whence } t_2 - t_1 = \frac{Q}{2\pi\lambda} \log_e \frac{r_2}{r_1} \tag{6}$$

If equation (6) is solved for Q we have

$$Q = \frac{(t_2 - t_1)\ 2\pi\lambda}{\log_e \frac{r_2}{r_1}} \tag{7}$$

This is for *unit length* of tube. But unit length has $2\pi r_2$ units of outside area, or unit outside area has $\frac{1}{2\pi r_2}$ units of length. So if Q' is the heat passing unit outside area,

$$Q' = \frac{Q}{2\pi r_2} = \frac{(t_2 - t_1)\ \lambda}{r_2 \log_e \frac{r_2}{r_1}} \tag{8}$$

This is not so convenient as the ordinary straight line formula. It is desirable to know what radius may be used so that equation (1) will give the same results as equation (7). If we let such radius be r_3, we find that when

$$r_3 = \frac{r_2 - r_1}{\log_e \frac{r_2}{r_1}} \tag{9}$$

and the area based on r_3 ($2\pi r_3$ for unit length of tube) is substituted in (1), then equations (1) and (7) give the same results. Equation (1) then reads for unit length of tube

$$Q = \frac{2\pi r_3\lambda\ (t_2 - t_1)}{r_2 - r_1} \tag{10}$$

Formulas containing natural logarithms are not convenient for engineering calculations. A simpler formula than (7) and (10) is desirable. Actually, it is found that *using in equation (1) the arithmetic mean of the outside and inside surfaces of a tube* will give a result within 10 per cent of that given by equations (8) or (9) for any ratio $\frac{r_2}{r_1}$ up to 3.2 or to within 1 per cent up to $\frac{r_2}{r_1} = 1.5$. Hence for all ordinary purposes the arithmetic mean is satisfactory and the working formula for a tube of length L becomes

$$Q = \frac{\pi\lambda L\,(r_2 + r_1)\,(t_2 - t_1)}{r_2 - r_1} \tag{11}$$

If, however, either the inner or the outer surface of a tube is used in formula (1), the result will differ from (8) by 1 per cent if the ratio $\frac{r_2}{r_1}$ is 1.02 or over; and by 10 per cent if $\frac{r_2}{r_1}$ is 1.24.

In practice the case often occurs of concentric layers of different conductivities. These cannot be calculated from equation (4) because in deriving equation (4) the *area* of each layer was assumed to be the same. If we have a tube consisting of concentric layers, whose conductances are K_1, K_2, K_3, and whose mean radii are r_1, r_2, r_3, (logarithmic means according to equation (9) if the layers are thick), then for unit length of tube the total resistance of the layers is $\frac{1}{2\pi r_1 K_1}$, $\frac{1}{2\pi r_2 K_2}$, etc. The total overall resistance per unit length of tube is the sum of these, and the total overall conductance per unit length of tube is the reciprocal of this sum. We usually deal in conductance per unit area, which in this case may be mean area, outside area, or inside area, whichever is convenient in design. The overall conductance per unit area; the *overall heat transfer coefficient*, is then

$$U = \frac{\frac{1}{2\pi r}}{\frac{1}{2\pi r_1 K_1} + \frac{1}{2\pi r_2 K_2} + \frac{1}{2\pi r_3 K_3}}$$

where r is the radius on which the heating surface is to be calculated. This may be simplified to

$$U = \frac{\frac{1}{r}}{\frac{1}{r_1 K_1} + \frac{1}{r_2 K_2} + \frac{1}{r_3 K_3}} \tag{12}$$

Problem. A steel tube is 0.10 inch thick, 1 inch inside diameter, with a hard rubber cover ⅛ inch thick (Fig. 2). Both on the outside of the rubber and on the inside of the steel are water films, each 0.01 inch thick. Heat is flowing from steam inside the tube to boiling water at 212° outside the tube. What must be the overall temperature drop to make heat flow at the rate of 200 B.t.u. per hour per square foot of outside surface? What will then be the overall coefficient U?

Formula (1) can be rearranged to read

$$t_2 - t_1 = \frac{QL}{A\lambda} \tag{13}$$

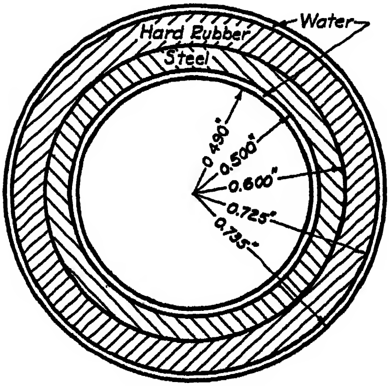

FIG. 2.

The outside surface of this tube per foot of length is $\frac{2\pi \times 0.725}{12}$ or 0.380 square foot. It will then be transmitting 200×0.380 or 76 B.t.u. per linear foot.

Drop across outside water film.

By formula (13) $t_2 - t_1 = \dfrac{76 \times (0.01/12)}{2\pi \times 0.418 \times 0.380} = 0.399°$

By formula (6) $t_2 - t_1 = \dfrac{76}{2\pi \times 0.418} \log_e \dfrac{0.735}{0.725} = 0.402°$

Drop across hard rubber.

The arithmetic mean surface is

$$\frac{\dfrac{(0.725 + 0.600)}{2} \times 2\pi}{12} = 0.347 \text{ square foot.}$$

By formula (11) (arithmetic mean surface)

$$t_2 - t_1 = \frac{76 \times (0.125/12)}{2\pi \times 0.021 \times 0.347} = 108.8°$$

By formula (6) (logarithmic mean surface)

$$t_2 - t_1 = \frac{76}{2\pi \times 0.021} \log_e \frac{0.725}{0.600} = 108.9°$$

By formula (13) (outside surface) $t_2 - t_1 = 99.5°$

Drop across steel.

By formula (11)	(arithmetic mean surface)	0.0853°
By formula (13)	(outside surface)	0.0782°
By formula (6)	(logarithmic mean surface)	0.0855°

Inside water film.

By formula (13) 0.579°
By formula (6) 0.573°

Then, if the temperature of the outside of the outer water film is 212°, we have

Temperature of boiling water	212.0°
Temperature of water-rubber surface	212.4°
Temperature of rubber-steel surface	321.3°
Temperature of steel-water surface	321.3°
Temperature of steam	321.9°

and the overall coefficient U is

$$\frac{200}{321.9 - 212.0} = 1.82 \text{ B.t.u. per square foot per hour.}$$

Evidently a rubber-covered tube could not be used under the above conditions, as destructive temperatures would exist in its inner layers.

A comparison of the results obtained by the different formulas shows that even in the rubber layer, the use of the arithmetic mean instead of the logarithmic mean causes a negligible error; and in the other layers even using the outside surface instead of the mean is not of much effect on the final result.

Mean Temperature Drop.

In the previous discussion, we have been considering the flow of heat across a wall whose surface temperature is uniform; or we may think of the preceding sections as applying to the flow of heat across a wall in one plane at right angles to the wall. In any actual piece of equipment, however, wall temperatures will vary from one part of the equipment to another. We may distinguish the following cases:

Case I. The temperature of the warmer side is uniform and that of the cooler side varies. An example would be a tube surrounded by steam at constant pressure with water flowing inside the tube.

Case II. The temperature of the cooler side is uniform and that of the warmer side varies. An example would be a boiler tube, containing water boiling under constant pressure, and surrounded by flue gases which cool as they impart heat to the tube.

Case III. The temperature of both sides varies in such a way that in passing from one end of the apparatus to the other, the temperature of one side rises and the other side falls.

Case IV. The same as Case III but both sides rise in temperature in the same direction. These two cases correspond, for instance, to the inner tube of a double pipe heat interchanger. In Case III the fluids flow in parallel current, and in Case IV in counter current.

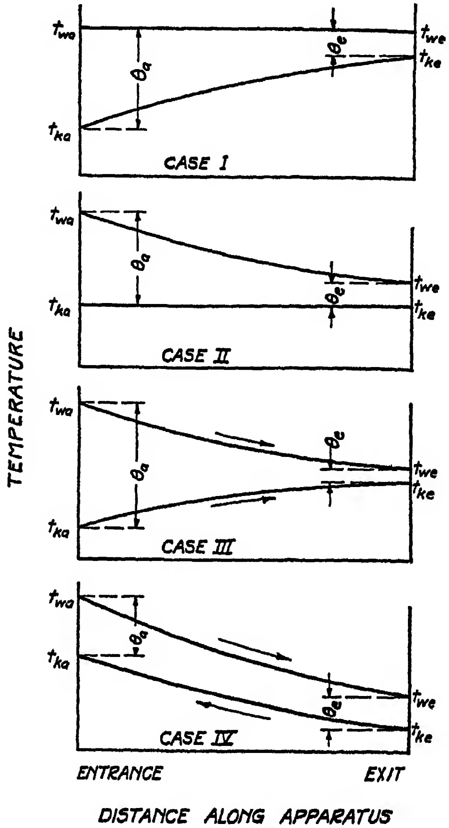
t_{we}
t_{we}
t_{ke}
Θ_a
Θ_e
t_{ka}
CASE I
t_{wa}
Θ_a
Θ_e
t_{we}
t_{ka}
t_{ke}
CASE II
t_{wa}
Θ_e
Θ_a
t_{we}
t_{ke}
t_{ka}
CASE III
t_{wa}
Θ_a
t_{ka}
Θ_e
t_{we}
t_{ke}
CASE IV
TEMPERATURE
ENTRANCE
EXIT
DISTANCE ALONG APPARATUS

FIG. 3.—Temperature distribution in heat exchange apparatus.

These cases may be illustrated by Figure 3. Here distance along the tube is indicated horizontally and temperatures vertically. The symbols are as follows:

t^{ka} = temperature of colder side at entrance of tube
t^{ke} = temperature of colder side at outlet of tube
t_{wa} = temperature of warmer side at entrance of tube
t_{we} = temperature of warmer side at outlet of tube
Θ_a = temperature drop at inlet of tube
Θ_e = temperature drop at outlet of tube

If we assume that heat transfer is proportional to temperature drop, the lines in the above figures cannot be straight except where the temperature does not change. If the lines are curved as shown, then the mean temperature drop cannot be the arithmetic mean. For all the above cases, the mean temperature drop (Θ_m) is given by the following expression:

$$\Theta_m = \frac{\Theta_a - \Theta_e}{\log_e \frac{\Theta_a}{\Theta_e}} \tag{14}$$

This may be derived as follows:

Let us consider a metal wall separating two fluids which are flowing along the wall and exchanging heat. (Figure 4.)

Let Θ = temperature difference at any point
Θ_a = temperature difference at inlet of tube
Subscript 2 refers to warmer fluid
Subscript 1 refers to colder fluid
t = temperature
S_1 = specific heat of colder fluid
S_2 = specific heat of warmer fluid
Q_1 = heat absorbed by colder fluid
Q_2 = heat absorbed by warmer fluid
W_1 = pounds of colder fluid flowing per unit of time
W_2 = pounds of warmer fluid flowing per unit of time

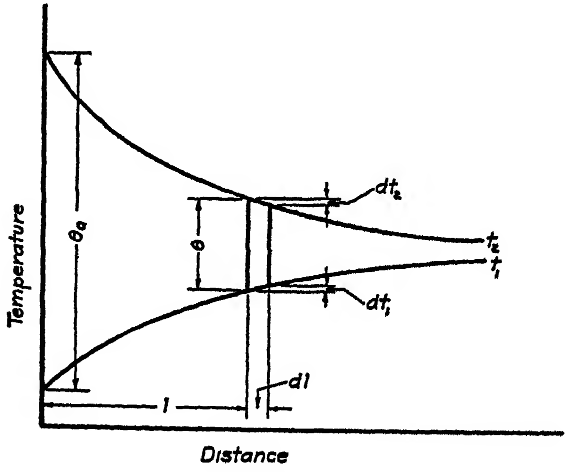

FIG. 4.

Consider a section of the tube of length dL:

$$dQ_1 = W_1S_1dt_1 \tag{15}$$

$$dQ_1 = -dQ_2$$

$$dQ_2 = W_2S_2dt_2 \tag{16}$$

$$W_1S_1dt_1 = -W_2S_2dt_2$$

$$\frac{dt_2}{dt_1} = -\frac{W_1S_1}{W_2S_2}$$

Subtracting unity from both sides

$$\frac{dt_2 - dt_1}{dt_1} = - \frac{(W_1S_1 + W_2S_2)}{W_2S_2} \tag{17}$$

$$t_2 - t_1 = \Theta$$

$$dt_2 - dt_1 = d\Theta$$

From (17) $dt_1 = - \dfrac{W_2S_2}{W_1S_1 + W_2S_2} d\Theta$ (18)

Substituting the value of dt_1 from (18) in (15)

$$dQ_1 = - \frac{W_1S_1W_2S_2}{W_1S_1 + W_2S_2} d\Theta = - A\, d\Theta \tag{19}$$

where $A = \dfrac{W_1S_1W_2S_2}{W_1S_1 + W_2S_2}$

$$Q_1 = - A \int_{\theta_a}^{\theta} d\Theta = A(\Theta_a - \Theta) \tag{20}$$

Q_1 is also equal to $- Q_2$.

Let H = square units of surface per unit length of tube.
K = overall heat transmission coefficient.

Then a section of the tube of length dL will have HdL units of surface, and in unit time there will pass

$$dQ_1 = HK\Theta dL$$

Taking the reciprocal of each side and multiplying by $d\Theta$

$$\frac{d\Theta}{dQ_1} = \frac{1}{HKdL} \times \frac{d\Theta}{\Theta} \tag{21}$$

But from (19) $\dfrac{d\Theta}{dQ_1} = - \dfrac{1}{A}$

Substituting in (21)

$$- \frac{1}{A} = \frac{1}{HKdL} \times \frac{d\Theta}{\Theta}$$

$$dL = - \frac{A}{HK} \frac{d\Theta}{\Theta}$$

$$L = - \frac{A}{HK} \int_{\theta_a}^{\theta} \frac{d\Theta}{\Theta} = - \frac{A}{HK} (\log_e \Theta_a - \log_e \Theta)$$

$$L = \frac{A}{HK} \log_e \frac{\Theta_a}{\Theta} \tag{22}$$

Now let us assume that there is some value for Θ (say Θ_m) such that

$$Q_1 = KHL\Theta_m$$

Whence

$$\Theta_m = \frac{Q_1}{KHL} \qquad (23)$$

Now if in (23) we substitute for Q_1 its value from equation (20) and for L its value in equation (22), we have

$$\Theta_m = \frac{A(\Theta_a - \Theta)}{HK \frac{A}{HK} \log_e \frac{\Theta_a}{\Theta}}$$

and

$$\Theta_m = \frac{\Theta_a - \Theta}{\log_e \frac{\Theta_a}{\Theta}}$$

Since this holds for all values of Θ, it holds where $\Theta = \Theta_e$, and we have then (14).

It should be noted that this is of the same general form as the equation for the mean radius of a tube (9). Therefore, the statements on page 15 hold here also; that is, if the arithmetic mean is used as the mean temperature drop, it will be in error by 10 per cent or more when $\frac{\Theta_a}{\Theta_e}$ is over 3.2 and by 1 per cent or more where $\frac{\Theta_a}{\Theta_e}$ is over 1.5.

Formulas like (14) containing natural logarithms are not very convenient for ordinary engineering work. Hausbrand[7] has shown that an approximation can be made in the form of

$$\Theta_m = N\,\Theta_a \qquad (24)$$

where N is a factor varying with $\frac{\Theta_e}{\Theta_a}$, and to be taken from the following table:

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[7] "Verdampfen, Kondensieren, und Kuhlen," 6th Ed., p. 9 (1918). The table given here does not correspond exactly with Hausbrand's table, his table as printed contained numerical errors

TABLE IV

$\frac{\Theta_e}{\Theta_a}$	N	Diff.	$\frac{\Theta_e}{\Theta_a}$	N	Diff.
0.0025	0.166		0.20	0.499	
		0.022			0.010
0.005	0.188		0.21	0.509	
		0.027			0.009
0.01	0.215		0.22	0.518	
		0.036			0.008
0.02	0.251		0.23	0.526	
		0.026			0.009
0.03	0.277		0.24	0.535	
		0.020			0.009
0.04	0.297		0.25	0.544	
		0.019			0.039
0.05	0.316		*0.30	0.583	
		0.018			0.037
0.06	0.334		0.35	0.620	
		0.016			0.035
0.07	0.350		0.40	0.655	
		0.015			0.035
0.08	0.365		0.45	0.690	
		0.014			0.034
0.09	0.379		0.50	0.724	
		0.013			0.032
0.10	0.392		0.55	0.756	
		0.013			0.030
0.11	0.405		0.60	0.786	
		0.012			0.029
0.12	0.417		†0.65	0.815	
		0.012			0.028
0.13	0.429		0.70	0.843	
		0.011			0.027
0.14	0.440		0.75	0.870	
		0.010			0.027
0.15	0.450		0.80	0.898	
		0.010			0.027
0.16	0.460		0.85	0.925	
		0.009			0.025
0.17	0.469		0.90	0.950	
		0.010			0.025
0.18	0.479		0.95	0.975	
		0.010			0.025
0.19	0.489		1.00	1.000	
		0.010			
0.20	0.499				

* For all values of Θ_e/Θ_a above this, the arithmetical mean will be 10 per cent or more in error.

† For all values of Θ_e/Θ_a below this, the arithmetical mean will be in error by less than 1 per cent.

Chapter 2.

General Theory of Heat Transmission—Radiation.

The fundamental law for the transfer of energy by radiation is the Stefan-Boltzmann law:

$$E = \sigma (T^4 - T_0^4) \tag{25}$$

where T is the absolute temperature of the hot body and T_0 the absolute temperature of the surroundings to which it radiates. σ is a factor determined by the nature of the hot body, and, to a lesser extent, by its temperature.

It should be noted that the above expression contains no terms referring to the temperature, pressure or composition of the fluid in the space between the two bodies. This is true so long as the medium is a completely transparent gas, but is not true if visible flames or smoke are present. Neither is radiation independent of the presence of transparent solids or liquids, for they may absorb heat without absorbing light, and thereby affect the amount of radiation passing between the bodies.

The Black Body.

If attempts are made to determine the numerical value of σ, the widest variations are found between different substances or different physical arrangements of the hot body. In particular it is found that those substances, such as polished metals, which reflect incident energy, are those which give the lowest values for σ. It is also noted that substances black in color and with rough or matte surfaces give the highest values. From this, there has been evolved the conception of the *ideal "black" body;* which is defined as one which absorbs all the radiation falling on it, and transmits and reflects none. It has been proven, both experimentally and theoretically, that the amount of energy a body radiates when it is the hotter of two is equal to the amount it absorbs when it is the cooler (all conditions being equal in the two cases). Hence, if the ideal black body absorbs more of the radiation incident on it than any other substance under the same conditions, it follows that it will also radiate more than any other substance. Also all black bodies will radiate the same amount of energy at the same temperature.

No actual physical substance is a black body, but substances of a matte surface and black color approach it. It can be shown that the inside of a closed space of uniform temperature is a black body, and any substance in that enclosure behaves like a black body. Based on this fact, experimental black bodies have been constructed and used for a study of radiation phenomena.

Distribution of Radiation.

In the above discussion the terms "radiation" and "energy" have been purposely used instead of "heat." The curves of Figure 5 show the distribution of energy in black body radiation at different temperatures. J is the amount of energy radiated, and μ is the wave length of that energy. Temperatures for the various curves are in degrees Fahrenheit. The space between $\mu = 0.4$ and $\mu = 0.75$ is approximately the range of the visible spectrum. It is obvious that at the temperatures indicated practically all energy radiated is heat energy. Even at excessive temperatures the amount of energy radiated other than as heat is small. The energy radiated as light waves or ultra-violet waves in an incandescent electric light is only about 4 per cent of the total.

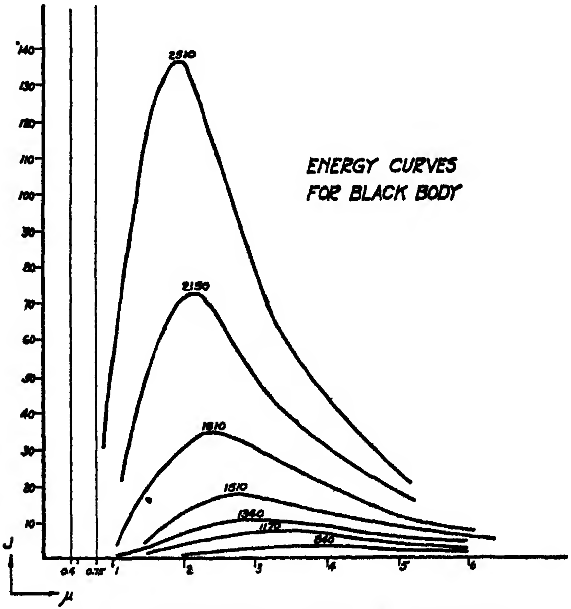

FIG. 5.—Distribution of black body radiation.

In general discussions of the laws of radiation, laws (such as Wien's and Planck's laws) are discussed which correlate wave lengths of maximum radiation with temperature, or energy of any particular wave-length with temperature. Since for our purposes, *all* the radiant energy may be considered as heat, such discussions are of no significance here.

Black Body Radiation.

The amount of energy (which, as shown above, we may consider as being all heat) radiated from a hot "black body" has been determined. The value usually taken is 1.28×10^{-12} gram-calories per square centimeter per second, where temperatures are measured in degrees absolute. Converted to engineering units, this becomes

1.62×10^{-9} B.t.u. per square foot per hour (temperatures measured in ° F. absolute).

There are certain further conditions which must be fulfilled before this value may be used. First, it can only be used where the radiation is received by another black body. For if the body receiving the radiation were not black, it would reflect some of the heat falling on it; and of this reflected heat, some would fall on the original radiating body and thereby decrease the total amount of energy radiated.

Second, the cooler body must be of such a size and in such a position as to subtend a spherical angle of 180° for the unit of surface of the hot body under consideration. This is ordinarily expressed by saying that the hot body must not be able to "see" anything but the cooler. Thus, for instance, the condition is satisfied in Figure 6b, and practically so for those parts of the hot body near the center in Figure 6a. It is not satisfied in Figures 6c and 6d. In Figure 6c, the element of the hot body indicated is so placed that the cold body subtends a solid angle of α. The rest of its "angle of vision" $(180° - \alpha)$ is occupied by other parts of the hot body itself. In this case, the element shown would radiate, not the amount called for by the value of σ for the black body, but $\frac{\sigma\alpha}{180}$ B.t.u. per square foot per degree F. absolute. In the case of Figure 6d, part of the "angle of vision" is occupied by other parts of the hot body, part by the cold body, and part by some more distant background.

Radiation Received by the Cooler Body.

It should be noted that this discussion is from the standpoint of a unit of surface of the hot body, and the energy received per unit of surface of the cooler body is not specified. This is illustrated by Figure 6b, where the amount of heat radiated by the indicated element of the hot body is all received by the cooler body, no matter which of the three suggested shapes or positions it has (provided it subtends an angle of 180° and is always at the same temperature). But the amount of heat received *per unit area* of the cooler body will vary greatly with the shape and position of the cooler body, and with the location of the particular element of its surface which is under consideration.

If we wish to base our calculations on the radiation received per unit area of the cooler body, then all the discussion of the previous section holds if the relations are reversed. The present discussion is limited to black bodies, so that condition is unaltered. Then one square foot of the cooler surface will receive 1.62×10^{-9} B.t.u. per square foot per degree F. absolute, if the hot body subtends a spherical angle of 180° with respect to the particular element of the cooler under consideration. It makes no difference what the size, shape, or distance of the hot body may be, provided only it fills the "angle of vision" of this element of the cooler body. In Figures 6a to 6d, the words "hot body" and "cold body" may be exchanged if we are considering the heat received by unit surface of the cold body.

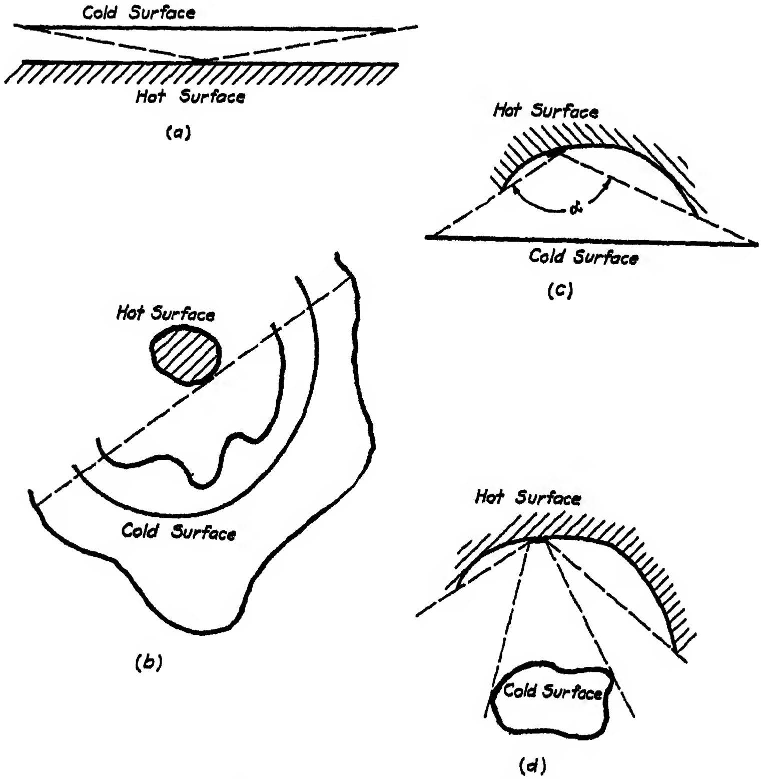

FIG. 6.—Effect of shape and distance on radiation.

As mentioned above, no actual physical substance is a black body, though some approach it closely. For the exchange of heat between dark, matte surfaces, the value of σ for a black body probably holds very closely. For such cases, with an "angle of vision" of 180 spherical degrees, the amount of heat actually exchanged per square foot is given in Figures 7 and 8.

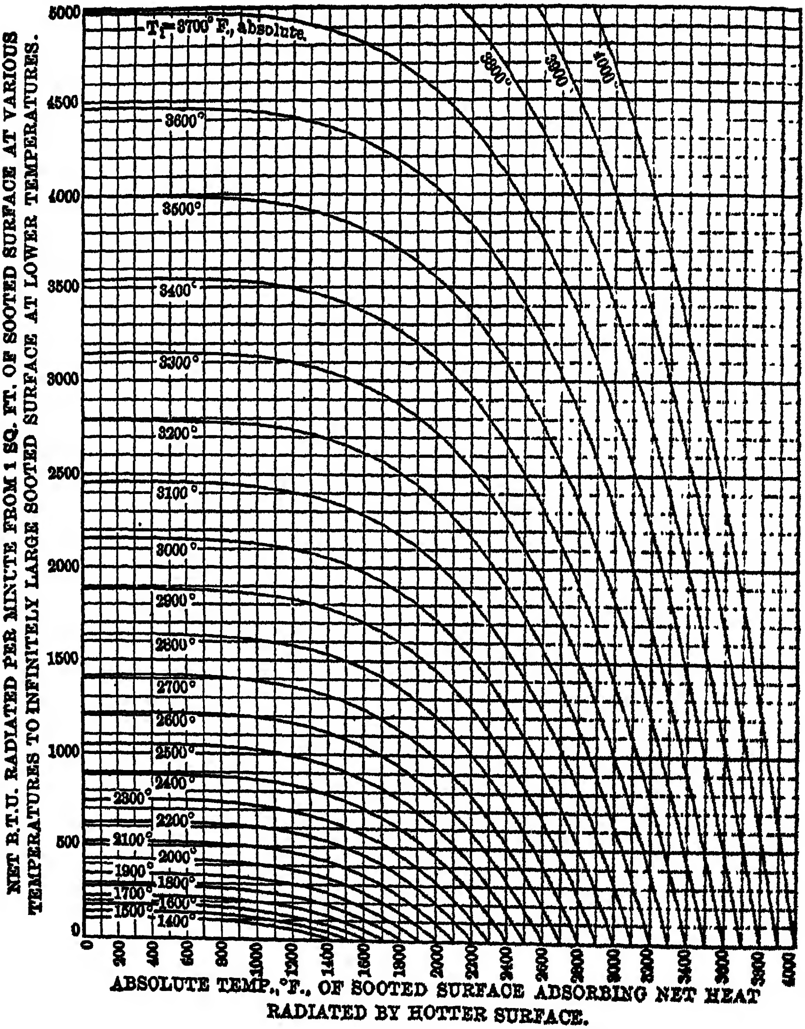

Fig. 7.—Heat transferred by black body radiation.

Non-black Body Radiation.

Wamsler[1] has determined the radiation constant for a number of materials. The material, in the form of a sphere with internal electric

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[1] *Mitt. Forschungsarbeiten*, No. 98 (1911).

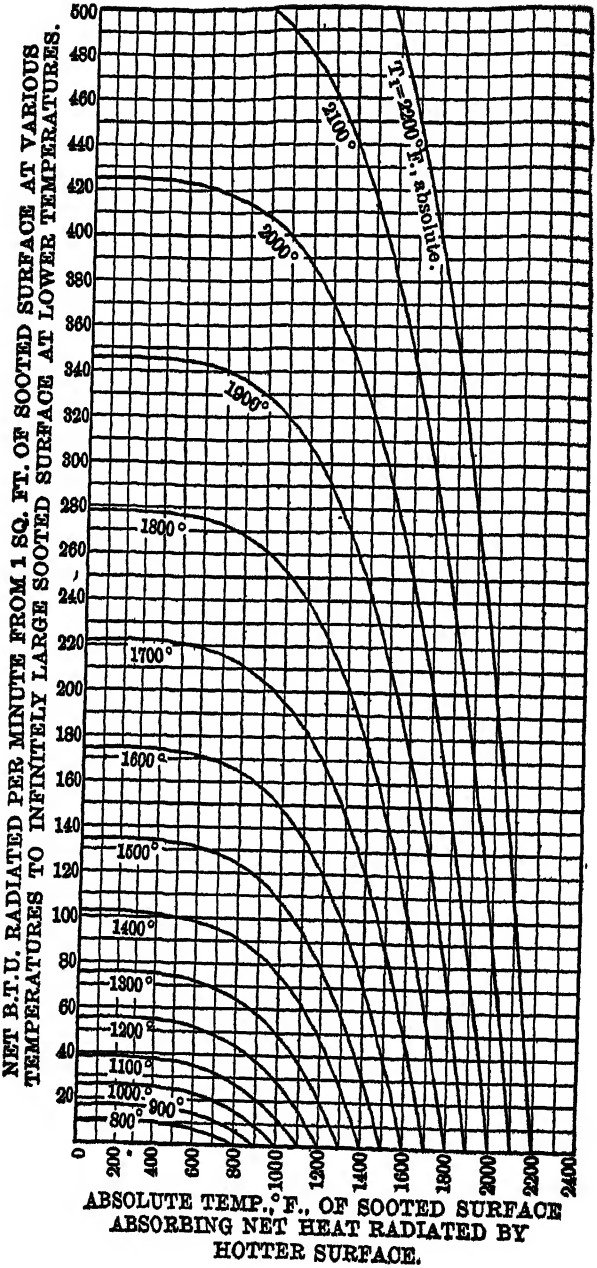

Fig. 8.—Heat transferred by black body radiation (lower part of Fig. 7 to an enlarged scale).

heating elements, was hung in the center of a larger sphere which was blackened on the inside and maintained at 0° C. Thus the receiving body was practically a black body, and the cooler everywhere filled the "angle of vision" of the hotter. Wamsler's determinations covered the temperature range from room temperature to 500-600° F., except for lime mortar, which was tested to 180° F. His results in B.t.u. per square foot per hour per degree F. absolute are as follows:

TABLE V

Radiation Constants

Material	$\sigma \times 10^9$
Lampblack	1.56
Brass (*matte*)	0.374
Copper	0.28
Sheet iron:	
Matte, oxidized	1.55
Highly polished	0.466
Zinc	0.343
Cast iron	1.58
Lime mortar	1.51

To this table should be added the value 1.34×10^{-9} for unglazed ceramic material given by Lewis.[2] The assumptions made in deriving this last figure are such that it has only a very approximate value.

These constants, like those for the black body, may be used only under certain conditions. First, the receiving body must be black. Second, it must totally fill the "angle of vision" of the hot body. Or, if the discussion is to be from the standpoint of the cooler, one square foot will absorb the amount of heat indicated if the radiator is a black body and if it fills the "angle of vision" of the cooler body.

If both the radiating and receiving bodies are not black, the case is still more complicated. Each body will be reflecting a part of the heat falling on it, and this will affect the exchange of heat. For the case where the cooler body completely surrounds the warmer, and the warmer has no re-entrant angles, Wamsler shows that the amount of energy radiated by the warmer body is

$$H = \frac{F_1 (T_1^4 - T_2^4)}{\frac{1}{\sigma_1} + \frac{1}{\sigma_2} + \frac{1}{\sigma}} \tag{25}$$

where F_1 is the area of the warmer body, T_1 its temperature and σ_1 its radiation constant, T_2 and σ_2 the temperature and radiation constants of the cooler body, and σ the black body constant.

Application to Design.

The above principles may be easily interpreted qualitatively and are a valuable guide in design. The following conclusions are worth noticing:

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[2] *J. Ind. Eng Chem.*, *7*, 410-4 (1915).

(1) The radiating surface should be maintained at as high a temperature as possible. Since radiation increases as the fourth power of the absolute temperature, a slight increase in the temperature of the warmer body has greatly exaggerated effect, while similar differences in the temperature of the cooler body make less difference, relatively.

(2) So far as radiation received by the cooler body is concerned, the hot body need only be large enough to fill its "angle of vision." Increasing the area of the warmer body (say, the brick setting of a fuel-fired pot) does not increase the amount of radiation after the "angle of vision" of the pot is filled, and does increase losses by radiation and conduction. However, it may sometimes be desirable to increase the brick surface to permit more complete transfer of heat from the hot flue-gases to the brick with consequent increase in wall temperature.

(3) In securing a large absorbing surface on the cold body, extensions of its surface which cannot "see" the hot body or which screen other parts of the cold surface, do not add to its rate of heat absorption. This is particularly true of corrugations, projecting pins or knobs, or other devices sometimes employed. Such projections may, by increasing area of contact or turbulence of flow, increase the heat absorbed directly from hot gases, but they cannot increase the heat absorbed by radiation from the setting. Extensions of the cold surface which can be made without going out of "sight" of hot surfaces will increase the heat absorbed.

Quantitatively, it is difficult to apply the laws and constants of radiation to actual design problems. The "blackness" of the surfaces involved is one uncertainty. The "angle of vision" of the cold body is another. In most cases in practice, the cold body will be able to "see" several different surfaces, all with different radiation constants and different temperatures. The angle each subtends will vary with every element of the cold surface considered. However, there is no reason for not applying actual calculations to those cases where the arrangement is simple enough to fulfil the conditions for the use of the constants.

Chapter 3.

Heat Transfer by Convection—General.

In Chapter 1 we have defined convection as the transfer of heat from a warmer to a cooler body by physical transfer of material. It becomes important, therefore, to consider the general phenomena of fluid[1] flow.

Critical Velocity.

The first idea which is of significance is that of critical velocity. When fluids flow at very low velocities, all the individual particles are flowing in parallel lines. A jet of colored water, introduced into the center of a tube in which clear water is flowing at a velocity below its critical velocity, will maintain itself throughout the tube as a separate colored line, which may waver a little but never loses its identity.

Now if the velocity of the water in such a tube is gradually increased, there will be a point at which the colored streak will suddenly disappear, leaving the whole mass of liquid colored. The fluid no longer flows in parallel lines, but by a series of eddies which result in a complete mixing of all parts. These two methods of flow are called, respectively, straight line or viscous, and sinuous or turbulent. The velocity at which the flow changes from straight-line to turbulent or vice versa is called the *critical velocity*. Two values are distinguished. When the flow of a fluid in the turbulent region is gradually decreased, it passes to viscous flow at a definite velocity called the lower critical velocity. When the experiment is reversed, and the velocity of a fluid in viscous flow is gradually increased, somewhat above the lower critical velocity a region of instability is reached, and the flow may pass over into turbulent flow at any point over quite a range of velocities. The highest velocity that may be reached in viscous flow is usually called the upper critical velocity, but it is not very definite. When this expression "critical velocity" is used, the lower value is ordinarily meant.

Critical velocity varies with viscosity, diameter, and specific gravity. In discussing friction losses in the flow of liquids in pipes, it has been shown[2] that f, the friction factor in Fanning's formula, may be expressed as a function of $\frac{Dvs}{z}$ where D is inside pipe diameter in inches, v is velocity in feet per second, s is specific gravity referred to water at 39° F., and z is absolute viscosity in centipoises. Wilson, McAdams, and Seltzer have plotted f over a wide range of conditions, and shown that $\frac{Dvs}{z} = 0.12$ at the lower critical velocity, and 0.2 to 0.4 at the upper critical velocity. Based on this, Table VI has been calculated.

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[1] The word "fluid" throughout this chapter is used where it is desired to include specifically both gases and liquids.

[2] Wilson, McAdams, and Seltzer, *J. Ind. Eng. Chem.*, *14*, 105-19 (1922).

TABLE VI

Lower Critical Velocity of Water (in Feet per Second)

Pipe Size Briggs Stand.	Temperature ° F. 32°	70°	120°	160°	212°
⅛	0.80	0.44	0.25	0.18	0.13
½	0.35	0.19	0.11	0.079	0.057
1	0.20	0.11	0.065	0.047	0.034
2	0.10	0.057	0.033	0.024	0.017
3	0.070	0.038	0.022	0.016	0.012
4	0.053	0.029	0.017	0.012	0.009
6	0.035	0.019	0.011	0.008	0.006

These values are based on commercially smooth pipe in lengths over 300 diameters. They are much lower than figures given in many reference works, because the latter quote experiments in glass or in very smooth brass tubes. Unusual roughness, burrs at joints, elbows or other fittings, and short lengths, all tend to lower the critical velocity.

The same formula serves for gases as well as liquids. Table VII shows the lower critical velocities for air and steam, with the same qualifications as Table VI.

TABLE VII

Lower Critical Velocity (in Feet per Second) of Air and Steam

Pipe Size Briggs Stand.	Temperature of Air, ° F. 32°	70°	160°	212°	Steam 212° F.
⅛	6.05	6.95	9.22	10.40	9.30
½	2.62	3.01	3.98	4.52	4.02
1	1.55	1.78	2.36	2.68	2.38
2	0.79	0.91	1.20	1.36	1.21
4	0.40	0.46	0.62	0.70	0.62
6	0.27	0.31	0.41	0.46	0.41

Distribution of Velocity.

Another conception regarding fluid flow which is useful in considering heat transfer is the fact that in either viscous or turbulent flow, velocity is not uniform across the pipe section. The curve of Figure 9 is from experiments by Stanton[5] and shows the general type of distribution curve usually found in turbulent flow. The average velocity is about 0.8 times the velocity at the center. In viscous flow there is a greater drop from center to wall, giving a more pointed curve than

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[5] *Proc. Royal Soc.*, Series A, *85*, 366 (1911).

TABLE VIII

VISCOSITY OF WATER[a]

(Centipoises)

Temperature ° F.	0°	1°	2°	3°	4°	5°	6°	7°	8°	9°
30................	....	....	1.794	1.759	1.726	1.693	1.662	1.631	1.602	1.573
40................	1.546	1.519	1.493	1.468	1.443	1.420	1.397	1.374	1.352	1.331
50................	1.310	1.290	1.270	1.251	1.232	1.213	1.196	1.178	1.161	1.145
60................	1.129	1.112	1.097	1.082	1.066	1.052	1.037	1.023	1.009	0.995
70................	0.982	0.969	0.956	0.943	0.931	0.919	0.907	0.895	0.884	0.872
80................	0.862	0.851	0.840	0.830	0.820	0.810	0.800	0.791	0.782	0.772
90................	0.764	0.755	0.746	0.737	0.729	0.721	0.713	0.705	0.697	0.690
100................	0.682	0.675	0.668	0.661	0.654	0.647	0.641	0.635	0.628	0.622
110................	0.616	0.609	0.603	0.597	0.591	0.586	0.581	0.575	0.570	0.565
120................	0.559	0.554	0.549	0.544	0.539	0.535	0.530	0.525	0.521	0.516
130................	0.511	0.507	0.502	0.498	0.494	0.490	0.486	0.482	0.478	0.474
140................	0.470	0.466	0.462	0.459	0.455	0.451	0.448	0.444	0.440	0.437
150................	0.433	0.430	0.427	0.423	0.420	0.417	0.413	0.410	0.407	0.404
160................	0.401	0.398	0.395	0.392	0.389	0.386	0.383	0.381	0.378	0.375
170................	0.373	0.370	0.367	0.365	0.362	0.359	0.357	0.354	0.352	0.350
180................	0.347	0.345	0.343	0.340	0.338	0.336	0.334	0.332	0.330	0.327
190................	0.325	0.323	0.321	0.319	0.317	0.315	0.313	0.311	0.309	0.307
200................	0.305	0.303	0.301	0.299	0.297	0.295	0.294	0.292	0.290	0.289
210................	0.287	0.285	0.284	0.282	0.280	0.279	0.277	0.275	0.274	0.273
220................	0.271	0.270	0.268	0.267	0.266	0.264	0.263	0.262	0.260	0.259

[a] Circular of the National Research Council, 1925. These values are accurate to 0.1 per cent up to 100° F., and to 0.5-1.0 per cent above this temperature.

TABLE IX

DENSITY AND SPECIFIC GRAVITY OF WATER[4]

Temperature ° F.	Density, Lbs. per Cu. Ft.	Specific Gravity	Temperature ° F.	Density, Lbs. per Cu. Ft.	Specific Gravity
32	62.41	.9999	140	61.38	.9832
35	62.41	.9999	145	61.28	.9818
40	62.41	1.0000	150	61.19	.9803
45	62.41	.9999	155	61.09	.9787
50	62.40	.9997	160	60.98	.9771
55	62.38	.9994	165	60.88	.9755
60	62.36	.9990	170	60.78	.9739
65	62.33	.9986	175	60.68	.9722
70	62.29	.9980	180	60.57	.9704
75	62.26	.9974	185	60.45	.9686
80	62.22	.9966	190	60.33	.9668
85	62.16	.9958	195	60.24	.9650
90	62.10	.9950	200	60.10	.9631
95	62.04	.9941	205	59.99	.9611
100	61.98	.9931	210	59.87	.9592
105	61.91	.9920	212	59.80	.9584
110	61.84	.9909	215	59.74	.9572
115	61.77	.9898	220	59.62	.9551
120	61.70	.9886	230	59.36	.9510
125	61.62	.9873	240	59.09	.9468
130	61.55	.9860	250	58.83	.9425
135	61.45	.9846	260	58.55	.9379

[4] Smithsonian Tables, 120 (1919).

Figure 9 and an average velocity about half the maximum. It will be noted that as the pipe wall is approached, the curve becomes nearly vertical. From various considerations, we are led to believe that a very thin layer next the wall actually does have zero velocity; that is, it is held to the wall so tightly that it cannot move along the pipe.

Transfer of Heat to Moving Fluids.

If a fluid be moving past a heated body (such as the wall of a pipe or other container in which it flows), it will receive heat by all three forms of heat transfer. It may absorb heat by simple conduction. But the thermal conductivities of all fluids are very low, and a thin layer has so much thermal resistance that no appreciable amount of heat could travel to a body of fluid by pure conduction. It may absorb heat radiated from the hot body. If the temperature of the hot body is so high as to cause appreciable radiation, the fluid in contact with such a wall can only be a gas and gases are, so far as we are concerned in this discussion, completely transparent to radiant heat. If the fluid is a liquid, it might absorb radiated heat, but liquids cannot ordinarily be exposed to wall temperatures high enough to transfer appreciable heat by this method. Hence in practice, fluids are heated entirely by convection, and the study of heat transfer by this means is correspondingly important.

The Film Concept.

According to the constants on page 11, Chapter 1, a plate of copper, one foot square and 0.1 inch thick with a drop of 1° F. between the two faces, should transfer at 212° F., $220 \times \frac{12}{0.1} = 26{,}400$ B.t.u. per hour. Actually, in practice, if we have steam on one side and boiling water on the other, we will find about 300-500 B.t.u. passing through one square foot per hour per ° F. If one side of the plate is in contact with a gas, the heat transferred will be of the order of 1/100 of these figures. Yet the fluids, in themselves, may not show any temperature gradient. In other words, this combination behaves as though there were some large resistance to the passage of heat at the surfaces where the fluids are in contact with the metal.

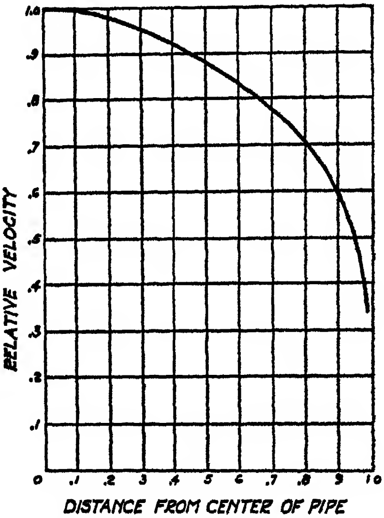

Fig. 9—Distribution of fluid velocity in turbulent flow.

An inspection of the curves for velocity distribution of Figure 9 and a consideration of the mechanism of fluid flow gives us a very useful working concept of heat transfer to fluids. In the first place, Figure 9 gives us a picture of stagnant or slow-moving layers of fluid next the metal. In the next place, the concept of turbulent flow, with particles of fluid moving in every direction, gives us the idea of heat transferred to a fluid only as this turbulence brings particles into contact with the hot layer, or tears off particles from the hot layer. Osborne Reynolds,[6] who was the first to discuss heat transfer from this point of view, thought of the process as a purely dynamic one, heat being imparted to the fluid only by the impact of molecules against the wall. The concept of a stagnant film of fluid is a very convenient one, however, and leads to the same ends. One thinks, then, of a fluid flowing over a heating surface, insulated from it by a stagnant layer, and heated only as it tears off and mixes into the main body particles from this layer which has been heated.

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[6] Collected Papers, Vol. I, pp. 81-5. This important paper has been largely copied in U. S. Bur. Mines, Bull. No. 18, p. 90, and in Royds, "Heat Transmission by Radiation, Conduction, and Convection," 1921, pp. 111-15.

In the light of this discussion, then, the temperature gradient between two fluids, separated by a metal wall, would be somewhat as illustrated in Figure 10. The total temperature drop, $t_1 - t_2$, is then made up of three parts: $t_1 - t_3$, the drop across the film on one side; $t_3 - t_4$, the drop across the wall; and $t_4 - t_2$, the drop across the second fluid film. And in strict analogy to fall of potential in electric circuits, the fall of temperature indicates quantitatively the magnitude of the resistance. This is saying in words what has been developed as equation (5), Chapter I.

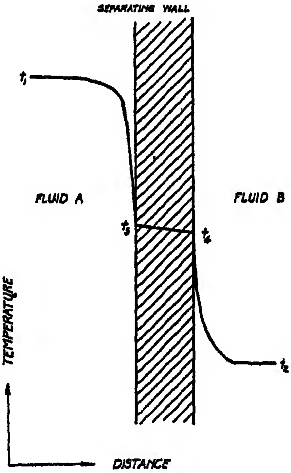

FIG. 10.—Temperature gradient through fluid films.

If we call K_1 the conductance of the film on one side, λ the conductivity of the wall, d the wall thickness, K_2 the conductance on the other side of the wall, and U the resulting overall conductance, we will have

$$\frac{1}{U} = \frac{1}{K_1} + \frac{d}{\lambda} + \frac{1}{K_2} \text{ and } U = \frac{1}{\frac{1}{K_1} + \frac{d}{\lambda} + \frac{1}{K_2}} \tag{26}$$

For thick-walled tubes, the more rigid form would be, from equation (12),

$$U = \frac{\frac{1}{r}}{\frac{1}{r_1K_1} + \frac{d}{r\lambda} + \frac{1}{r_2K_2}} \qquad (27)$$

where r is the mean radius of the metal of the tube, and U is therefore heat transfer per square foot mean tube surface. U is usually called the "overall heat transfer coefficient" and is the final form in which results are desired for engineering work.

It is obvious that the various factors which affect either K_1 or K_2 will have an indirect effect on U. But their effect is obscure enough so that any study of U where conditions are varied gives too incomplete a result to be of much value. Useful results will only be obtained when K_1 or K_2 are studied directly. Unfortunately, only a beginning has been made in such work. We will discuss these results as far as they go, but it will be necessary in much engineering work, to use empirical values for U for some time to come.

Effect of Velocity—Reynolds' Equation.

Reynolds, in his fundamental paper, says: "The heat carried off by air, or any fluid, from a surface, is proportional to the internal diffusion of the fluid at and near the surface; i.e., is proportional to the rate at which particles and molecules pass backwards and forwards from the surface to any given depth within the fluid. Thus, if AB be the surface and ab an ideal line in the fluid parallel to AB, then the heat carried off from the surface in a given time will be proportional to the number of molecules which in that time pass from ab to AB; that is, for a given difference of temperature between the fluid and the surface.

"Now this rate of diffusion has been shown from various considerations to depend on two things:

1. The natural internal diffusion of the fluid when at rest.
2. The eddies caused by visible motion which mixes the fluid up and continually brings fresh particles into contact with the surface.

"The first of these causes is independent of the velocity of the fluid and, if it be a gas, is independent of its density, so that it may be said to depend only on the nature of the fluid. The second cause, the effect of eddies, arises entirely from the motion of the fluid, and is proportional both to the density of the fluid, if gas, and the velocity with which it flows past the surface.

"The combined effect of these two causes may be expressed in a formula as follows:

$$H = A\Theta + B\rho v\Theta \qquad (28)$$

where Θ is the difference of temperature between the surface and the fluid, ρ is the density of the fluid, v its velocity, A and B constants depending on the nature of the fluid, and H the heat transmitted per unit area of the surface per unit time."

If we express ρ as pounds per cubic foot, and v as feet per second, ρv is the weight of fluid per square foot of cross-section of fluid path per second. If the area of the path is *a* square feet, and the total weight of fluid passing per second is w, then $\frac{w}{a} = \rho v$. Further $\frac{H}{\Theta} = K$. Substituting these in equation (28) and using subscripts 1 and 2 to distinguish between the fluids on the two sides of the metal wall, we have

$$\begin{aligned} K_1 &= A_1 + B_1\frac{w_1}{a_1} \\ K_2 &= A_2 + B_2\frac{w_2}{a_2} \end{aligned} \qquad (29)$$

This is our first approximation in attempts to find the actual value of K_1 and K_2. The numerical values will be discussed under the various special cases as they arise.

Chapter 4.

Heat Transfer between Solids and Gases.

We are considering here, strictly speaking, the *film* coefficient only (K_1). But we find, after surveying the whole field, that if, on the other side of the solid, we have condensing steam or liquid, the value of K_2 is ordinarily so large ($\frac{1}{K_2}$ so small) that the gas film resistance is practically the total resistance. For any metal heating surface ordinarily used, $\frac{d}{\lambda}$ in equation (26) is quite negligible. Hence we may take $U = K_1$ for a first approximation, although there is some work in which K_1 has actually been studied.

Experimental Results.

Reynolds' equation (29) has been generally accepted and forms the starting point for most investigations. It is a straight-line equation, and hence we would expect K_1 (or U in this case) to increase in direct proportion to $\frac{w_1}{a_1}$. Plots of the relation between K_1 and $\frac{w_1}{a_1}$ as determined by various investigators are shown in Figure 11. Here $\frac{w_1}{a_1}$ is, as defined above, pounds of gas per square foot of path per second, and K_1 is B.t.u. passing from metal to air (or vice versa) per square foot per degree F. mean temperature difference. Because most of the data was available on the basis of B.t.u. per second, this has been used in Figure 11 instead of B.t.u. per hour. A description of the various experiments follows:

Curve 1. Nusselt[1] passed air, at pressures from 1.4 to 214 pounds per square inch (gage) through a brass tube 0.87″ I.D., 1.02″ O.D., which was surrounded by a steam jacket kept at about 215°. The temperature of the metal wall on the steam side was measured at one point with a thermocouple. Air temperatures were measured with a platinum resistance grid covering the whole cross-section of the tube and shoved along inside the tube for measurements at different points. The maximum length of tube used (between air temperature readings) was 30″,

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[1] *Mitt. Forschungsarbeiten*, No. 89 (1910). Converted to English units by Royds, "Heat Transmission by Radiation, Conduction, and Convection," 125 (1921).

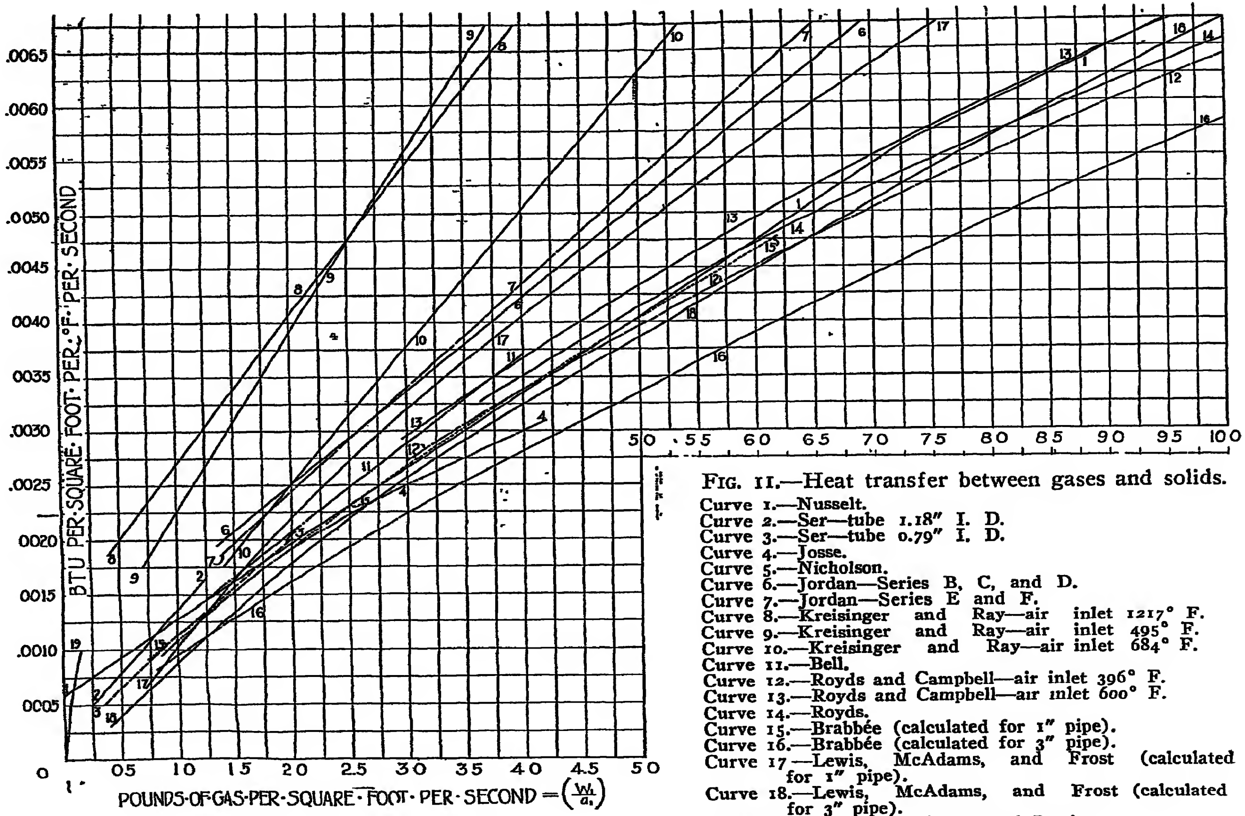

FIG. 11.—Heat transfer between gases and solids.

Curve 1.—Nusselt.
Curve 2.—Ser—tube 1.18" I. D.
Curve 3.—Ser—tube 0.79" I. D.
Curve 4.—Josse.
Curve 5.—Nicholson.
Curve 6.—Jordan—Series B, C, and D.
Curve 7.—Jordan—Series E and F.
Curve 8.—Kreisinger and Ray—air inlet 1217° F.
Curve 9.—Kreisinger and Ray—air inlet 495° F.
Curve 10.—Kreisinger and Ray—air inlet 684° F.
Curve 11.—Bell.
Curve 12.—Royds and Campbell—air inlet 396° F.
Curve 13.—Royds and Campbell—air inlet 600° F.
Curve 14.—Royds.
Curve 15.—Brabbée (calculated for 1" pipe).
Curve 16.—Brabbée (calculated for 3" pipe).
Curve 17.—Lewis, McAdams, and Frost (calculated for 1" pipe).
Curve 18.—Lewis, McAdams, and Frost (calculated for 3" pipe).
Curve 19.—Snyder, Robinson, and Lewis.

and the average was about 24″. Air inlet temperatures ranged from 70° to 140°, outlet temperatures from 100° to 185° F. The "shape factor"[2] was 0 218″. Variations in pressure apparently had no effect on K_1, as all the points lie very close to the line plotted. Nusselt expresses his results by the following equation:

$$K_1 = 26.9 \frac{\lambda \text{ wall}}{d^{0\ 214}} \left(\frac{vC_p}{\lambda}\right)^{0\ 786} \tag{30}$$

where λ wall is the thermal conductivity of the gas at the temperature of the wall (in B.t.u. per square foot per foot per degree F. per hour), d the inside diameter of the pipe in inches, v the velocity of the gas in feet per second, C_p the specific heat of the gas in B.t.u. per *cubic foot* at mean gas temperature, and λ the thermal conductivity of the gas at mean gas temperature. In this formula the effect of λ, d, and C_p is determined on theoretical grounds only. (Nusselt used only one tube diameter.) In a tube approximately 1″ in diameter, with steam at about 215° outside it, Nusselt's results give

$$K_1 = b \left(\frac{w_1}{a_1}\right)^{0\ 786} \tag{31}$$

where b has the following values:

Gas	b
Air	4.11
CO_2	3.40
Coal gas	11.00
Superheated steam	7.68

Curve 2. Ser[3] passed air inside a copper tube of 1.18″ I.D., heated by warm water. Temperatures were measured with mercury thermometers. The data are reported in such a way that the recalculation is of questionable value. The shape factor is 0.296″.

Curve 3. This is the same as Curve 2 except that the tube is 0.79″ I.D., and the shape factor 0.20″.

Curve 4. Josse[4] passed air at pressures from atmospheric to 0.5″ (absolute) through a tube 0.905″ I.D. and 52″ long, heated with steam. When his experiments are calculated on the basis of $\frac{w_1}{a_1}$, the points all fall very closely on the line plotted, irrespective of the absolute pressure of the air. The shape factor is 0.226″.

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[2] The "shape factor" is $\frac{\text{area of gas path}}{\text{perimeter of heating surface}}$. In a tube of any cross section it is the same as the hydraulic radius; in an annular space it differs from hydraulic radius in that the denominator includes only the perimeter of the heating surface, not the whole perimeter of the path. For a circular tube the shape factor is half the radius.

[3] "Traite de Physique Industrielle," 1888. Converted to English units by Royds, p 151.

[4] *Engineering, 86,* 802-6 (1908). Converted to English units by Royds, p. 155.

Curve 5. Nicholson[5] passed air at 60 pounds per square inch and at 100° to 160° F. down the annular space between a tube 0.516″ O.D. and one 0.898″ I.D. Cold water was passed down the central tube. The shape factor of the annular space was 0.26″. His rates of flow were very large (his lowest value of $\frac{w_1}{a_1}$ being 25.5). His points seem to determine a straight line, the extrapolation of which to values of $\frac{w_1}{a_1}$ in the range of Figure 11 fall outside the area reproduced.

Curves 6 and 7. Jordan[6] passed air at atmospheric pressure and temperatures from 250° to 600° F. down a copper tube surrounded by a water jacket. He measured the temperature of the tube at 4 points with thermocouples, and inlet and outlet air with mercury thermometers. On plotting tube temperatures against length, he got a slightly curved line, which he assumed to be a straight line for simplicity. By so doing he obtained an expression

$$M\,L = \log_e \frac{T_1 - t_1 - \frac{c}{M}}{T_2 - t_2 - \frac{c}{M}} \qquad (32)$$

where $M = \frac{K_1 \pi d_1}{K w_1}$, d_1 is the inside diameter of the air tube, K is the specific heat of air, L is tube length, T_1 and T_2 are air temperatures at inlet and outlet, t_1 and t_2 tube temperatures at inlet and outlet, and c is the temperature gradient along the tube $\left(\frac{t_1 - t_2}{L}\right)$. To find K_1 it was necessary to substitute experimental values in equation (32), solve for M, substitute in the expression for M and solve for K_1. Then to find the mean temperature difference between air and metal, it was necessary to divide the total heat transferred per square foot per hour by K_1. This round-about method leaves considerable opportunity for error. It may be shown as a corollary of equation (14) that the tube temperature t_2 at a point distant L from one end should be

$$t_2 = t_0 e^{-pL} \qquad (33)$$

where t_0 is the temperature at the end of the tube from which distance is measured. If (33) holds, the mean temperature difference between metal and either fluid is given by equation (14). Jordan does not give data on his tube temperatures, but does plot one curve to illustrate the distribution of temperature. This curve as printed is of the form of (33) as closely as his plot can be read.[7] Hence Jordan's assumption of a straight line tube temperature introduces unnecessary complications. How much error is introduced cannot be determined, as data

[5] *Junior Inst. of Eng. 1908.* Reported in Royds, pp. 155-6.

[6] *Proc. Inst. Mech. Eng. 1909-II,* 1317-57.

[7] Dr. T. Running, University of Michigan, has worked this out in a private communication.

are not given which would permit recalculating his results. The assumption which he makes causes his values for Θ_m to be too small, and hence his values for K_1 are too high. A much more serious error may have been introduced by using mercury thermometers to measure air temperatures, but his description of his apparatus is too incomplete to discuss this in detail.

Jordan ran five series of experiments in which the shape factor was varied. In series D, E, and F, the air path was a simple tube; in series B and C the air path was made annular by filling the center of the tube. Shape factors were: Series B, 0.171″; Series C, 0.26″; Series D, 0.492″; Series E, 0.1265″; Series F, 0.309″. Points in the various series spread considerably, but Curve 6 is a rough mean of Series B, C, and D, while Curve 7 for Series E and F shows higher values of K_1 at higher rates of flow. The spreading of experimental points away from the mean curves is very noticeable as compared with the work of Josse and Nusselt. Jordan expresses his results in terms of Reynolds' equation (29), $\frac{H}{\Theta} = K_1 = A + B\frac{w_1}{a_1}$. He finds, for all his experiments, $A = 5.25$. B he finds to vary with hydraulic radius and with wall temperature, so that his final formula is

$$K_1 = 5.25 + \left[1.82 - 1.62r + 0.00595\frac{T+\Theta}{2}\right]\frac{w_1}{a_1} \qquad (34)$$

where r = hydraulic radius in inches, T = mean air temperature, Θ = mean temperature of air side of tube.

Curves 8, 9, and 10. Kreisinger and Ray[8] heated air by an electric heater and drew it at atmospheric pressure through small boilers with tubes 0.230″ I.D. for No. 2 and 0.175″ I.D. for No. 3. Curve 8 shows their results for boiler No. 2 with an entrance air temperature of 1217° F., Curve 9 is for the same boiler with air at 405° F., and Curve 10 for boiler No. 3 with 684° inlet temperature. Their points fall very close to the lines plotted. The shape factor for No. 2 is 0.057″ and for No. 3, 0.044″.

Curve 11. Bell[9] passed gases from a gas-fired furnace through a 2-inch copper pipe with individual water jackets along its length. Mean temperature differences varied from 400° to 2000° F., though the curve shown in Figure 11 is for $\Theta_m = 400°$ only. For higher temperature differences the curves are still straight, but lie higher on the plot and have a steeper slope. The shape factor is 0.50″.

Curves 12 and 13. Royds and Campbell[10] passed hot air at atmospheric pressure through a steel tube 0.994″ I.D. (shape factor 0.248″), which was surrounded by a water jacket. Air temperatures were taken with mercury thermometers. Curve 12 is for an air inlet temperature of 396° F., and Curve 13 for 600° F. The points fall very close to the curves.

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[8] U. S. Bur. Mines, Bull. 18, 1912 Data recalculated by Royds, pp. 166-7.
[9] Mech. Eng. Section, Int. Eng. Cong., 1915 Plotted by Royds, p. 172.
[10] *Trans. Inst. Eng. Shipb. Scot*, 55, 153-207 (1912).

Curve 14. Royds[11] did further work in the same apparatus which is reproduced as Curve 14. The plotted points fall fairly close to the curve. Air pressure was varied here from 15 to 55 pounds absolute, higher pressures tending to give higher results. Air temperature was held at about 500° F.

Curves 15 and 16. Brabbée[12] passed air through clean iron pipes 0.85″ to 4.7″ I.D. and 39″ long, heated by steam jacket or by hot water. He neglected $\frac{d}{\lambda}$ and K_2, calling K_1 the total resistance. His experiments are expressed by the equation

$$K_1 = 4.056 \frac{(\gamma v)^{0.79}}{d^{0.16}} \qquad (35)$$

where γ = density of air in pounds per cubic foot, v = velocity of air in feet per second, and d = I.D. of pipe in inches. Curve 15 has been calculated from this formula for air at atmospheric pressure and 70° F. inside 1-inch pipe, and Curve 15 for the same conditions in 3-inch pipe.

Curves 17 and 18. Lewis, McAdams, and Frost[13] suggest, for heat transfer between a metal and *any* gas,

$$K_1 = \frac{0.88\left(\frac{w_1}{a_1}\right)^{0.8} C_p S^{0.2} \sqrt{T}}{M^{0.3}} \qquad (36)$$

where C_p is the specific heat of the gas at constant pressure in B.t.u. per pound, S is the reciprocal of the shape factor in feet, T is the mean temperature of the gas in Fahrenheit degrees, and M is the molecular weight of the gas. Curve 17 has been calculated for air at a mean temperature of 70° F., inside a 1-inch pipe; and Curve 18 is the same in a 3-inch pipe.

Curve 19. Snyder, Robinson, and Lewis[14] passed air through a ½-inch brass pipe 27 inches long (shape factor 0.125″), heated electrically, and determined K_1 as a function of $\frac{w_1}{a_1}$. Their rates of flow were so small $\left(\frac{w_1}{a_1} \text{ less than } 0.015\right)$ that their results are of little significance.

The first impression from Figure 11 is that the results fall in a fairly well defined group except for Curves 6, 7, 8, 9, 10, 17, and 19. The original data for Curve 5 (which falls entirely outside the part of the plot reproduced) have not been examined, and it is difficult to state the cause of the error. Curves 6 and 7 (Jordan) are possibly high, due to the errors explained on page 43 Curves 8, 9, and 10 were run with air hotter than in most of the other experiments. This may cause K_1 to be high, but at high temperatures the measurement of a gas temperature is very difficult and an error here may have been the cause of the high results. Curve 17 is from the Massachusetts Institute of Technology formula, in which great weight has evidently been given to Jordan's work. Curve 19 is so far from the others as to be altogether without significance.

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[11] *Trans. Inst. Eng. Shipb. Scot., 58,* 155-211 (1915).

[12] *Mitt. Prüfanstalt für Heizungs- u. Luftungsanlagen, Heft 3,* 1910, quoted in Hausbrand, p. 114.

[13] Bull. Mass. Inst. Tech., *57,* No. 33 (contribution from Dept. of Chem. Eng., Serial No. 4), 1922.

[14] *J. Ind. Eng. Chem., 8,* 515-17 (1916).

Effect of Various Factors on K_1, Velocity.

Evidently the value of K_1 is not a straight-line function of $\frac{w_1}{a_1}$ as Reynolds' equation requires. At high gas speeds there is enough more turbulence to cause the layers of low velocity near the wall to decrease in thickness sufficiently to affect the results. Probably $\frac{w_1}{a_1}$ should enter as an exponential function with a power somewhat less than 1 (0.786, Nusselt; 0 79, Brabbée; 0.80, Lewis, McAdams, and Frost).

Shape Factor.

It is reasonable to suppose that an increase in the shape factor of the gas path should decrease the value of K_1. The larger the tube, the smaller the per cent of the total number of particles passing a given cross-section that will strike the wall in a given length of time. The diameter of the tube (which is 4 times the shape factor for a path of circular cross-section) is introduced into the denominator by Nusselt with the exponent 0.214 (determined from theoretical consideration only), by Brabbée with the exponent 0.16, by Lewis, McAdams, and Frost with the exponent 0.2 (their S is the reciprocal of shape factor as used here), and it is introduced by Jordan as a term decreasing the value of B in Reynolds' equation. A study of the data of Figure 11 fails to show any correspondence between values for K_1 and shape factor, and we are therefore not justified in introducing it into the formula at the present.

Film Temperature.

Since the principal resistance to the flow of heat is a layer of gas through which heat passes by conduction, and since the thermal conductivity of gases increases with temperature, we should expect to find the value of K_1 increasing with film temperature. What the relation of film temperature is to gas temperature is not so clear. Nusselt uses the temperature of the wall, Jordan the mean of wall and gas. There is, however, no general conclusion to be drawn from Figure 11 as regards this effect.

Nature of Gas.

It is evident that with a given degree of turbulence, if a particle of one gas when it strikes a metal wall is able (by virtue of its greater density or greater specific heat) to absorb more heat than a similar particle of another gas, the first gas will show a larger value of K_1 (other things being equal). Nusselt introduces the specific heat directly, and the molecular weight in the denominator. Very little work has been done on other gases than air; and the simpler formula of Nusselt (31) seems to be the best we have.

Working Value of Gas Film Coefficient.

In the light of the above discussion, a curve has been drawn, expressing the weight of opinion in Figure 11, whose equation is

$$K_1 = 4.32 \left(\frac{w_1}{a_1}\right)^{0.75} \tag{37}$$

and which is tabulated for different values of $\frac{w_1}{a_1}$ in Table X. This is to be used for air, flowing inside tubes of from ½ inch to 2 inches diameter at velocities above the critical, and at air temperatures not over 400° F. For gases other than air the work of Nusselt, as expressed in equation (31), is to be preferred. The coefficients to be used in (37) are then

CO_2	3.58
Coal gas	11.60
Superheated steam	8.06

For larger pipes the value of K_1 will be somewhat less, for higher temperatures it will be somewhat greater than given in the tables. We are not as yet justified in being more specific. All the work on which equation (37) is based was done with steam or hot water or boiling water on the other side of the wall from the gas. In such cases the values of K_2 and $\frac{d}{\lambda}$ in equation (26) are so large as compared with K_1 that they may be neglected. Hence, in plotting Figure 11, no distinction has been made between those cases where K_1 has been determined as such, and those cases where only an overall coefficient has been determined. Only in case there is a gas on both sides of the wall, or the wall is unusually thick or an unusually poor conductor, need the values of equation (37) be used as film coefficients.

Problem. 50,000 cubic feet of air per hour (measured at 30″ and 70° F.) are to be cooled from 220° F. to 70° F. by passing through a cooler consisting of 1″ standard iron pipe in a cylindrical shell, the air being inside the pipes and cooling water outside. The water is to enter at 40° F. and leave at 150° F. The cooler is to be so proportioned that the exit velocity of the air is to be about 20 feet per second. What should be the heating surface?

Number of pipe needed in parallel.

$$\frac{50{,}000}{3{,}600} = 13.89 \text{ cubic feet gas per second.}$$

$$\frac{13.89}{20} = 0.694 \text{ square feet, total area of gas path.}$$

The internal cross-sectional area of standard 1-inch pipe is 0.864 square inch.

$$\frac{0.694 \times 144}{0.864} = 115.8 \text{ or } \mathit{116\ pipe.}$$

Heat transfer coefficient. If 116 pipe are in parallel, the actual exit velocity will be within less than 1 per cent of the desired velocity. At 20 linear feet per second at 70° F., the velocity is 20 cubic feet per square foot cross-section. Since 1 cubic foot of air under these conditions weighs 0.0751 pounds,

$$\frac{w_1}{a_1} = 20 \times 0.0751 = 1.5020$$

From Table X, $K_1 = K =$ *5 86 B.t.u. per square foot per hour per ° F.*

Mean temperature drop. Since the exit temperature of the water is higher than the exit temperature of the air, the apparatus is obviously to be operated in counter current.

$$\Theta_a = 220 - 150 = 70. \quad \Theta_e = 70 - 40 = 30.$$

$$\frac{\Theta_e}{\Theta_a} = 0.428. \quad \text{From Table IV, } N = 0.678.$$

$$\Theta_m = 70 \times 0.678 = 47.5°.$$

(Note that the arithmetic mean, 50°, is 5.0 per cent in error.)

Total heat transferred. The mean specific heat of air in this range is 0.2375 B.t.u. per pound.

$$50{,}000 \times 0.0751 \times 0.2375 \times (220 - 70) = 1{,}338{,}000 \text{ B.t.u. per hour.}$$

Proportions of heater.

$$1{,}338{,}000 = 5.86 \times 47.5 \times \text{H.S.}$$

Heating surface = 480 square feet.

One-inch pipe has 1 square foot mean surface for 3.27 running feet. Therefore, for every 3.27 feet length of this heater there are 116 square feet of heating surface.

$$\frac{480}{116} \times 3.27 = \mathit{13.5\ feet\ length\ of\ tubes\ needed.}$$

TABLE X

$$K = 4.32\left(\frac{w_1}{a_1}\right)^{0.75}$$

$\frac{w_1}{a_1}$	K_1	$\frac{w_1}{a_1}$	K_1
0.10	0.77	5.0	14.4
0.20	1.29	5.5	15.6
0.30	1.75	6.0	16.6
0.40	2.17	6.5	17.6
0.50	2.57	7.0	18.6
0.75	3.48	7.5	19.6
1.00	4.32	8.0	20.6
1.25	5.10	8.5	21.6
1.50	5.86	9.0	22.5
1.75	6.57	9.5	23.4
2.00	7.27	10.0	24.3
2.25	7.94	10.5	25.2
2.50	8.59	11.0	26.1
2.75	9.23	11.5	27.0
3.00	9.85	12.0	27.9
3.25	10.5	12.5	28.7
3.50	11.1	13.0	29.6
3.75	11.6	13.5	30.5
4.00	12.2	14.0	31.3
4.25	12.8	14.5	32.1
4.50	13.3	15.0	33.0
4.75	13.9		

Transfer of Heat from Flue Gases.

In Figure 11 Curve 11 is the only curve determined on flue gases, and it does not deviate enough from work done on air to be significant in the present state of our knowledge. Nusselt, in the formula quoted on page 42, gives a coefficient for CO_2 in equation (31) 17 per cent less than for air. Hence a flue gas containing not over 15 per cent CO_2 should give a result not over 2.5 per cent less than air, which is an accuracy entirely outside that of our present information. Of course, in practice, if heat were to be transferred from flue gases to metal or vice versa a larger factor of safety would have to be used than when air is considered, because of the coating of soot and dirt apt to form on the heating surface.

It is beyond the scope of this work to go into the transfer of heat in steam boilers. It is often desirable, however, to use waste heat in flue gases for evaporation and heating. For this type of design, it is probable that the factors given above for air will be directly applicable.

Economizers.

One type of apparatus for transferring heat to non-boiling liquids is the economizer, which has been fairly well standardized by mechanical engineers as a steam boiler accessory. Due to the corrosive action of SO_2 when flue gases are cooled below their dew-point, it is usually considered necessary to make economizer tubes of cast iron. One well-known design consists of banks of cast iron tubes, arranged to take water in parallel, and over which the stack gases flow at right angles. It is usually necessary to provide scrapers to keep down deposits of soot and dust. Details of design of economizers may be found in standard works on power plant engineering.

There are, in the literature, a considerable number of economizer tests, but in practically no cases have gas quantities been determined, and therefore it is very difficult to orient them Hausbrand[15] gives a considerable collection from German sources, and calculates U and Θ_m for each. The figures seem to indicate a tendency to increase with increasing Θ, but this is probably due to the fact that higher values for Θ mean high gas temperatures, which in turn mean high rates of combustion or high drafts. These involve higher values of $\frac{w_1}{a_1}$, whence the increase. In general, all that can be said is that for commercial designs in practice one may expect U = 1.0 to 1.6 B.t.u. per square foot per hour per ° F.

Superheated Steam.

A special case of heat transfer to or from gases in which the chemical engineer is especially interested is that involved in handling superheated steam. Poensgen[16] investigated the film coefficient to superheated steam and proposed the following formula:

$$K_1 = 0.024 \frac{p^{1.082}\, v^{0.802}}{d^{0.164} \times 10^{(0.00085t - 0.03)}} \tag{38}$$

Hausbrand[17] has calculated a number of values from this equation and from Nusselt's equation (30), from which it appears that Nusselt's formula gives decidedly higher results than Poensgen's. Nusselt's constant of 7.68 for equation (31) is calculated from Poensgen's data. Equation (31) is simpler, and is perhaps safe for use at present. We may then say that for the transfer of heat from metal to superheated steam, with the steam inside small tubes, the coefficient is

$$K_1 = 8.06 \left(\frac{w_1}{a_1}\right)^{0.8} \tag{39}$$

Hausbrand[18] quotes a number of tests on superheaters in which the total overall coefficient is determined. Values for U vary from 2 to 6 B.t.u. per square foot per ° F. per hour. They seem to be greater with greater gas velocities, and to change little with steam velocities. He states that general experience shows that 1 square foot of superheater surface will superheat 5 to 10, or even in some cases 12 pounds steam per hour through 200° to 350°, when the flue gases are at 850° to 1000° F.

---

[15] "Verdampfen, Kondensieren, und Kuhlen," 6th Ed., 26-30 (1918).

[16] *Gesundheits-Ing.*, *39*, 257-61 (1916).

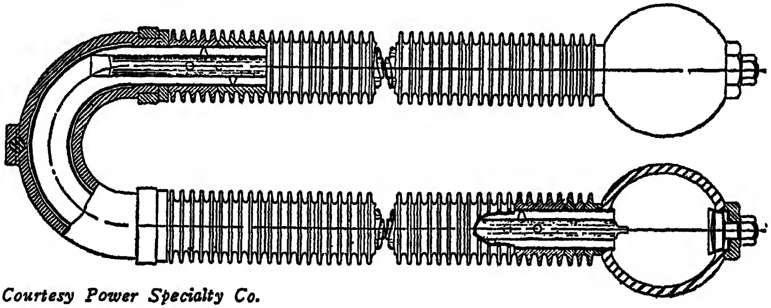
[17] *Loc. cit.*, pp 133-4.

[18] *Loc. cit.*, pp. 48-53.

It should be noted that in superheating steam by flue gases, the partial coefficient K_1 from equation (38) or (39) is *not* equal to the overall coefficient U, as has been assumed for other cases mentioned above. In considering heat transfer to or from superheated steam, equation (26) must be used.

Use of Superheated Steam.[19]

The design of superheaters for power purposes is a subject which belongs to the mechanical engineer. In chemical engineering practice, it may sometimes be convenient to use superheated steam for heating purposes. Superheated steam can be used at moderate or low pressures to give a working temperature drop that would involve very high pressures if reached with saturated steam, or danger of overheating the product if reached with direct fire. The transfer of heat from superheated steam to metal is, however, of the same order of magnitude as from any permanent gas to metal, and this is very much less than with saturated steam. Further, it is not practical to use superheated steam in a closed coil or jacket, for only a small per cent of the total heat in the steam is present as superheat. Hence, if superheated steam containing 5 per cent, say, of its available heat as superheat is introduced into a closed coil, and only condensate removed from the coil, 95 per cent of the heat would have to be transmitted with a temperature drop corresponding to saturated steam. But under the conditions assumed, this temperature drop is supposedly too low; hence the coil would become filled with saturated steam and be inoperative. Superheated steam can only be used where the coil can exhaust nearly saturated steam into the air or to low-pressure mains.



Courtesy Power Specialty Co.

FIG. 12.—Steam superheater tubes.

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[19] See Foster, *Chem. Met. Eng.*, *26*, 458-62 (1922).

Superheaters.

Steam will ordinarily be superheated by the use of direct fire, either in the boiler in which it is generated, or in a separately fired heater. The temperatures to which the superheating coils are subjected are so high as to make the use of unprotected steel tubes questionable. The Power Specialty Company make a superheater tube whose construction is shown in Figure 12. A seamless steel tube is covered with special cast iron protecting covers, shrunk on to the tube. In the center of the tube is a filler so that the steam flows in a narrow annular path. These tubes may be exposed to very high temperatures.

Chapter 5.

Heat Transfer between Solids and Liquids.

This phase of the general problem of heat transfer has received little attention and is not in a satisfactory state. The general phenomena are the same as discussed by Reynolds (see pages 38 and 39, Chapter 3, and his equation (28) should apply to this case also. So long as we confine the discussion to water, ρ may be considered constant and combined with B, so that for water we should have

$$\frac{H}{\Theta} = A + B'v \qquad (40)$$

In other words, the flow of heat between metal and water should increase in direct proportion to water velocity.

Reynolds' point of view is essentially that the transfer of heat from a metal wall to a fluid occurs by the same physical process that causes friction; namely, the impact of fluid particles against the wall. He later developed[1] the idea that friction between a fluid and the wall of the passage through which it flowed was some function of $\frac{Dv\rho}{z}$ where D is diameter, ρ is density, and z viscosity. Therefore, this expression should be related also to the flow of heat from a metal to a flowing fluid.

General Theory.

From another point of view the same function has been developed.[2] Let us assume that heat transfer is a function of film thickness, and that this film thickness in turn is a function of pipe diameter and of the velocity, viscosity, density, and thermal conductivity of the fluid. In other words

$$K = f\left(\frac{1}{\text{Thickness}}\right) = \phi(D^x v^y \rho^m z^n \lambda^p) \qquad (41)$$

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[1] Collected Papers, Vol. II, 51-105; *Phil. Trans., 174* (III), 935-82 (1883).

[2] Porter, *Eng., 107,* 125-6, 140-2 (1919); Nusselt, *Mitt. Forschungsarbeiten, 89,* 1-38 (1910); McAdams and Frost, *J. Ind. Eng. Chem., 14,* 1101-5 (1922); Technical Records of Explosives Supply, British Ministry of Munitions, Report No. 9.

Now the dimensions of both sides of the equation must be the same. By equating the dimensions of the two sides, it is found that the relation must be

$$\frac{KD}{\lambda} = f\left(\frac{Dv\rho}{z}\right)^n \tag{42}$$

Due to the difficulty of determining film resistances (since this necessarily involves determining tube wall temperature), most of the work so far done is very fragmentary, and the results are contradictory and difficult to interpret. Fluid velocity is the only variable that has been generally considered.

Experimental Work.

English and Donkin[3] studied the flow of heat between a short thick-walled metal tube and water flowing inside the tube. Wall temperatures were measured by drilling holes parallel to the axis of the tube, filling them with mercury, and inserting slender thermometers. Their apparatus was so small (the tube was only 2.75 inches long) and their temperature measurements so uncertain, that their results need not be given.

Stanton[4] passed water through a copper tube, and also around it in a jacket. The rise in temperature of the water inside the tubes was kept small (not over 16° F.) and the quantities of water inside and outside the tube were the same. Hence the temperature of the metal itself was probably uniform from end to end; and was measured by measuring the length of the tube. But the small rise in temperature introduced a chance of considerable error in measuring the total heat passed. He used three tubes, all about 18 inches long, and 0.548 inch, 0.421 inch, and 0.290 inch inside diameter.

His final conclusion is that the transfer of heat from metal to water is given by the following equation:

$$K_1 = 228\,\frac{p^{0.15}v^{0.85}}{d^{0.15}}\,(1 + 0.0024T_0)\,(1 + 0.00675t) \tag{43}$$

where v is velocity of the water in feet per second, d is inside pipe diameter in inches, T_0 is wall temperature (° F.), and t is mean water temperature. p is a function from Poiseuille's formula for pipe friction, and is equal to $(1 + 0.0336t_c + 0.000221t_c^2)^{1}$, where t_c is mean water temperature in degrees C. This function has the following values when T is measured in degrees F.:

T (° F.)	$p^{0.15}$	T (° F.)	$p^{0.15}$
40	0.98	70	0.92
50	0.95	80	0.90
60	0.94	90	0.88

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[3] *Proc. Inst. Mech. Eng.*, *1896*, 501-35.

[4] *Phil. Trans.*, *190*(A), 67-88 (1897). Abstracted very fully in U. S. Bur. Mines, Bull. 18.

T (° F.)	$p^{0.15}$	T (° F.)	$p^{0.15}$
100	0.87	160	0.80
110	0.86	170	0.79
120	0.84	180	0.78
130	0.83	190	0.77
140	0.82	200	0.76
150	0.81	212	0.75

Austin[5] worked with water or oil in rectangular compartments. They were separated by a cast iron wall about 1.2 inches thick, in which were thermocouples at three different distances from one face. Oil in one chamber was heated electrically. The total quantity of heat passing through the wall was calculated from the thermal conductivity of cast iron and the observed temperature gradient in the wall. He expressed his results, not as coefficients, but as a thickness of iron corresponding to the temperature drop between metal surface and water ($t_4 - t_2$, Fig. 10).

Clement and Garland[6] passed water inside a seamless steel tube 0.985 inch I.D., 6 feet 7¼ inches long, heated by steam, in a jacket 10 inches in diameter. The steam-surface temperature of the tube was taken with a thermocouple, and the temperature of the water surface calculated. Their results are very irregular, but are expressed directly as coefficients. They did not attempt to derive any general formula for K.

Soennecken[7] passed water down through tubes 6 feet 3½ inches long, heated by water in a jacket. Quantities of water inside and outside the tube were equal, as in Stanton's work, and tube temperature was measured by measuring the length of the tube. He used a brass tube 0.67 inch I.D., a steel tube 0.67 inch I.D., and another 1.10 inches I.D. His experiments seem to have been carried out very carefully, but experimental data are not given.

He derived the following formulas:

For brass tubes:

$$K_1 = 153\frac{v^{0.9}}{d^{0.1}}(1 + 0.0104T_0) \qquad (44)$$

For iron tubes:

$$K_1 = 148\frac{v^{0.7}}{d^{0.1}}(1 + 0.0104T_0) \qquad (45)$$

where v is water velocity in feet per second, d is inside diameter of pipe in inches, and T_0 is the temperature of the water side of the tube wall (° F.).

Webster[8] passed water through a copper tube 0.50 inch I.D., 30 inches long, heated by steam in a jacket. He measured tube temperatures with a thermocouple, protected by a steel sheath, which was brazed to the tube, extended across the steam jacket, and passed through a stuffing box. The steam jacket was only 7/16 inch wide, so that this mass of metal in the thermocouple sheath undoubtedly disturbed temperature conditions at the point of measurement. He did not calculate coefficients nor derive a general expression for water film coefficients, but gave full experimental data.

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[5] *Z. Ver. deut. Zuckerind., 54,* 635 (1904); *Z. Ver. deut. Ing., 1902,* 1890.

[6] Univ. of Ill. Eng. Exp. Sta. Bull. 40 (1909).

[7] *Mitt. Forschungsarbeiten,* No. 108-109, 33-78 (1911).

[8] *Trans. Inst. Eng. Shipb. Scot., 57,* 58-79 (1913).

Jordan[9] calculated coefficients on the water side of his tube, but the errors discussed in connection with his results on gas film coefficients hold here also.

Lewis, McAdams, and Frost[10] suggest the following equation for the film coefficient between any liquid and a metal surface:

$$K_1 = 712\frac{CKv^{0.8}}{zm^{0.17}} \tag{46}$$

where C is a cleanness and roughness coefficient, to be taken as 1 for smooth new pipe (copper, brass, cold-drawn seamless steel) and 0.5 for ordinary iron pipe, K is the thermal conductivity of the liquid, v its velocity in feet per second, m the hydraulic radius in feet, and z is the viscosity in centipoises. They say that this equation has been checked a few times on oils and agrees fairly well. In the case of water this reduces to

$$K_1 = 200\frac{v^{0.8}}{z} \tag{47}$$

McAdams and Frost published a later work,[11] in which they develop equation (39) to the form:

$$K_1 = \frac{23.3\lambda}{D}\left(\frac{Dv\rho}{z}\right)^{0.706} \tag{48}$$

They plot certain selected determinations from Webster, Clement and Garland, and unpublished theses, which confirm their equation very closely.

Webster's work comes the closest to their plot. Clement and Garland, working in an iron tube, got lower results (as might be expected considering the possibilities of rust). Jordan's work is very much higher than this curve, but Jordan's resistance on the gas side of his tube was so high that the temperature drop on the water side is very small and hence may be in error. Stanton's work is the only really careful work that deviates greatly from this equation.

Working Formulas.

All attempts at classifying the experimental work available have failed to yield any simple relation. This relation of equation (48) rests on sound theoretical grounds. It must be remembered that surface conditions are apt to play a large part in determining a coefficient. Probably these values are the safest now available, but the following points must be borne in mind:

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[9] *Loc. cit.*
[10] Bull. Mass. Inst. Tech., *57*, No. 33 (1922).
[11] *J. Ind. Eng. Chem.*, *14*, 1101-5 (1922).

1. This formula is based on experimental work done in clean copper or brass tubes only.
2. It has not been tested on tubes outside the range of 1 inch to 3 inches in diameter.
3. The authors say that a few points have been determined with oils (to cover a wide range of viscosities and conductivities), but such points are very few in number.
4. The precision of neither this experimental work nor the experimental work on heat transfer through gas films is enough to warrant a difference in exponents. The probability is that both should be the same. Whether 0.75 or 0.80 is correct cannot now be decided, but it is of small consequence in commercial design.
5. It has recently been suggested[12] that at about 80° C. (175° F.) there is a considerable change in conditions in the film, due to a change in solubility of dissolved gases, that quite vitiates relations which have been developed for its resistance at temperatures below this.

We may conclude, then, that for heat transfer between liquids inside clean copper or brass pipes and the metal wall, the film coefficient is given by

$$K_1 = \frac{23.3\lambda}{D}\left(\frac{Dv\rho}{z}\right)^{0.8} \tag{49}$$

but in developing this equation, its authors have used mixtures of units so that care must be taken in the application of this formula.

K_1 = film coefficient in B.t.u. per square foot per ° F. per hour.

λ = thermal conductivity of liquid at *mean film* temperature in B.t.u. per square foot per foot per ° F.

D = inside diameter of pipe in *inches*.

v = velocity of liquid in *feet* per second.

ρ = density in pounds per cubic foot at *mean temperature* of the whole column of liquid.

z = viscosity of liquid in centipoises at *mean film* temperature.

For liquids in channels other than circular pipes, no direct evidence is available. Probably heat transfer is affected here, as distinguished from circular pipes, only by the shape factor (note 2, Chapter 4). Since for circular pipes the shape factor is 0.25D, for channels other than circular, four times the shape factor should probably be used in place of D. For liquids in tanks or in heater shells outside tubes, or in any other case where shape factor of path and velocity of liquid cannot be determined directly, we have no way whatever to calculate the rate of heat transfer.

[12] Anon., *Engineering*, *116*, 1-3 (1923).

Double Pipe Heaters.

These are the simplest means of transferring heat from liquids to liquids through metal walls. They have an advantage in that calculations may be made for them with some pretense at precision. The fittings for one common type are shown in Figure 13. These are made only for 1¼-inch pipe inside 2-inch pipe, with standard screwed joints. By using standard ammonia flanges, similar fittings may also be had to take 2½-inch and 1¼-inch pipe, 3-inch and 2-inch, and 4-inch and 3-inch pipe.

For tubular heaters, the designs discussed in the next chapter may be used.

Another type of heat transfer apparatus is one in which the liquid to be cooled is pumped inside a series of pipes, and water is allowed to trickle down the outside of the stack, being cooled by evaporation. This stack may consist of standard pipe in standard return bends, or special welded constructions may be used. There are no published experimental studies on this type of cooler. Nusselt[13] has made an elaborate mathematical study of these devices, but his treatment requires so many simplifying assumptions that its final validity is questionable.

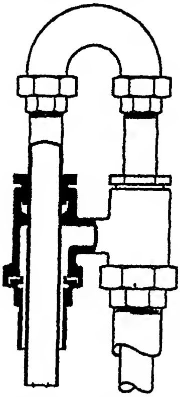

FIG 13.—Double pipe coil fittings.

Problem.

A double pipe cooler is made up of standard 1¼-inch pipe inside standard 2-inch pipe, using fittings as in Figure 13. The effective length per pass is 20 feet. In the inside pipe flows 850 gallons per hour of saturated salt brine, entering at 60° F. Hot condensed water at 220° is available for the annular space, and it is cooled to 120° F. Assume 2 per cent of the total heat given up by the hot water is lost by radiation, and that the hot water velocity may not be over 5 feet per second. How hot will the brine be heated, how many stacks of coolers will be needed, and how high will each stack be?

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[13] *Z. Ver. deut. Ing.*, *67*, 206-10 (1923).

From standard pipe tables we find:

	I. D.	O. D.	Inside cross-sect. area	Outside cross-sect. area
1¼-inch pipe	1.380″	1.660″	1.495 sq. in.	2.164 sq. in.
2 -inch pipe	2.067″	2.375″	3.355 sq. in.	4.430 sq. in.

Number of stacks.

$$\frac{850 \times 231}{3600} \times \frac{1}{1.495\mathrm{N}} = \text{velocity of brine in inches per second,}$$

where N is the number of stacks in parallel.

$$\frac{850 \times 231}{3600 \times 1.495 \times \mathrm{N}} = \frac{36.5}{\mathrm{N}}.$$

Three feet per second is a reasonable velocity; hence one stack is sufficient. The actual velocity will be 3.04 feet per second.

Temperature of Exit Brine.

The specific gravity of saturated salt brine at 60° F. is 1.202, which gives a weight of 75.0 pounds per cubic foot. The weight of brine entering per hour is

$$\frac{850 \times 231}{1728} \times 75.0 = 8510 \text{ pounds.}$$

The area of the annular space is 1.191 square inches. At an inlet temperature of 220° F. the hot water will weigh 59.7 pounds per cubic foot; and at an inlet velocity of 5 feet per second, its weight per hour will be

vel. [illegible]

$$\frac{3600 \times 1.191 \times 60}{1728} \times 59.7 = 8890 \text{ pounds.}$$

The specific heat of saturated salt brine is about 0.785, the specific heat of the hot water will be practically 1.00, the temperature drop of the hot water is 100°; so that with 2 per cent allowed for radiation, the useful heat given up will be

$$8890 \times 100 \times 1.00 \times 0.98$$

and the rise in temperature of the brine will be

$$\frac{8890 \times 98}{8510 \times 0.785} = 130° \text{ F.}$$

The exit brine temperature will then be 60 + 130 = *190° F.*

Mean Film Temperatures.

In passing through the apparatus, neither fluid changes in temperature linearly; hence the mean fluid temperatures will be the logarithmic mean of their inlet and outlet temperatures. That is

$$t_m = \frac{t_a - t_e}{\log_e \dfrac{t_a}{t_e}}$$

Using the method on page 23, Chapter 1, we have

for water—$t_a = 220°$, $t_e = 120°$, $\frac{t_e}{t_a} = 0.545$, factor $= 0.750$, $t_m = 165°$.

for brine—$t_a = 190°$, $t_e = 60°$, $\frac{t_e}{t_a} = 0.316$, factor $= 0.596$, $t_m = 113°$.

We do not yet know the film resistances; hence we do not know the temperature drop across the films. But if we assume that the film resistances are approximately equal, and that the drop across the metal is negligible, then the mean metal temperature is 139°, the mean brine film is 126°, and the mean water film is 152°.

Film Coefficient—Water Side.

Use equation (49). For D, we use four times the shape factor, which is (cross-sectional area of path)/(perimeter of heating surface). The mean circumference of 1¼-inch pipe is 4.775 inches. Hence, for the constants in equation (49) we have

$$D = \frac{4 \times 1.191}{4.775} = 1.00$$

$v = 5$

ρ (at entrance) $= 59.7$

z (at 152°) $= 0.426$

λ (at 152°) $= 0.384$

$$K_1 = \frac{23.3 \times 0.384}{1.00}\left(\frac{1.00 \times 5.0 \times 59.7}{0.426}\right)^{0.8} = 1690$$

Film Coefficient—Brine Side.

The thermal conductivity and viscosity of salt brines are not known; hence we will have to take the corresponding figures for water. This gives a value for the conductivity which is probably too low, and one for the viscosity also probably too low.

$D = 1.380$ $\qquad z = 0.529$

$v = 3.04$ $\qquad \lambda = 0.371$

$\rho = 75.0$

$$K_2 = \frac{23.3 \times 0.371}{1.380}\left(\frac{1.380 \times 3.04 \times 75.0}{0.529}\right)^{0.8} = 1040$$

Overall Heat Transfer Coefficient.

Since the films have different areas, equation (27) must be used.

$$U = \frac{\frac{1}{r}}{\frac{1}{r_1K_1} + \frac{d}{r\lambda} + \frac{1}{r_2K_2}}$$

where r is the mean radius of the pipe. This will then be the coefficient per square foot of mean pipe area.

$d = 0.140'' = 0.0117$ foot
$\lambda = 35$
$r_1 = 1.660''$, $r_2 = 1.380''$, $r = 1.52''$ (arithmetic mean)

For safety, considering probable rust or scale on the surfaces, use two-thirds of the calculated values of K_1 and K_2. Then

$$U = \frac{\frac{1}{1.52}}{\frac{1}{1.66 \times 1125} + \frac{0.0117}{1.52 \times 35} + \frac{1}{1.38 \times 690}} = 364$$

Number of Passes Needed.

The mean heating surface per pass is $\frac{4.775 \times 12}{144} \times 20 = 7.96$ square feet. The mean temperature drop is the logarithmic mean of (220 — 190) and (120 — 60).

$\Theta_a = 60$, $\Theta_e = 30$, $\frac{\Theta_e}{\Theta_a} = 0.500$, factor $= 0.724$, mean $= 43.4°$.

Then the heat transferred per pass is $7.96 \times 364 \times 43.4$, and the number of passes needed is

$$\frac{8890 \times 98}{7.96 \times 364 \times 43.4} = 6.9 \text{ or, say, } 7 \textit{ passes.}$$

It is interesting to note that if equation (26) (which does not take into consideration the curvature of the tube) is used, the overall coefficient comes out 374. This is an error of only 2.75 per cent, and our knowledge of the coefficients is not nearly accurate enough to warrant such precision. Hence the simpler equation (26) may be used except for very thick walled tubes.

Chapter 6.

Heat Transfer between Solids and Condensing Vapors (Especially Steam).

This phase of heat transfer has proven very difficult to analyze, from both the theoretical and the experimental aspects. The difficulties arise from the large number of variables which are here operative. If we consider the mechanism by which a particle of steam condenses and gives up its heat to the tube, we see that there are two separate processes which must be distinguished. The steam must give up its heat to a layer of condensate, and this in turn must transmit the heat to the metal wall. Assuming that there is no resistance to be overcome in transferring the heat from a particle of steam to the outer layer of condensate, we must consider the factors affecting

1. The impact of the steam particles upon the liquid film.
2. The thickness of the liquid film.
3. The transfer of heat through the film.

As discussed in Chapter 3, the *impacts of steam particles against the liquid film* are a function of the degree of turbulence of steam flow. Hence this again should be a function of $\frac{Dv\rho}{z}$, where D is the diameter, and v, ρ, and z are the velocity, density, and viscosity of the steam. If we could measure the rate of heat flow between steam and the outer surface of the condensate film, by determining the total heat passing and also determining the temperature of this outer layer, we might find that it was a simple power function of $\frac{Dv\rho}{z}$. Obviously, this is not possible, and hence our results are all obscured by the drop in temperature across the liquid film.

The significance of the terms of this factor are all obvious except v. The mean linear velocity of the steam through the tube is not the only velocity to be considered. Thus if A cubic feet of steam enter the tube, and B cubic feet of steam are vented, the mean steam velocity is usually considered to be the logarithmic mean of A and B. But if, in the process, on each square foot of heating surface, X cubic feet of steam are being condensed per minute, this causes a velocity of approach of X feet per minute. This is usually small compared to the linear velocity of the steam, but, in certain cases, it may be a factor in the number of impacts. For steam inside a tube, both D and v are fairly definite, but many cases arise in which they are not; such as the case of a tubular heater with steam outside the tubes, steam in the calandria of a vertical tube evaporator, etc.

Non-condensable Gases.

The presence of non-condensable gases in steam (or whatever vapor is being used) is a very important factor. Many workers have recognized the possibility of this variable affecting their results, but none has attempted to measure it or control it. It is of sufficient importance to warrant further discussion.

All steam contains air, brought into the system in solution in the boiler feed water. To eliminate corrosion there is a tendency in central stations at present to install apparatus where all boiler feed, both returns and make-up, is de-aerated by boiling under vacuum. This is only done at present in a few of the larger plants, but it is a tendency to be encouraged and one which will undoubtedly show considerable development in the near future.

Consider a boiler fed with raw water saturated with air at 70° F. Assume that all this air goes into the boiler (i.e., that none is removed by feed heaters), that it is all boiled off to go into the steam, and that 5 per cent of the feed water is removed, air-free, as blow-off. One cubic foot of water at 70° F. dissolves about 0.019 cubic foot of air, which corresponds to about 0.53 cubic inch air per pound H_2O. Hence, under the above assumptions, one pound of steam would contain 0.56 cubic inch of air. This is, of course, the extreme case, and is merely intended to show the upper limit of possible concentrations. It will be seen that, even at high pressures, this is a very small amount of air by volume in the steam itself, but it may result in considerable volumes of air accumulating in spaces where steam is condensed. For instance, a steam coil condensing 100 pounds of steam per hour would accumulate, one might think, 56 cubic inches of air in that time, enough (at atmospheric pressures) to fill a 2-inch pipe for a distance of 17 inches.

The problem is never so simple as this in practice. The air never collects in one part of the coil conveniently free from steam. In the first place, it must be saturated with water vapor at the temperature of the condensed water. If, for instance, the steam space of the coil were at atmospheric pressure and the condensed water were taken off at, say, 200° F., the vapor pressure of water would be about 600 mm., leaving only 160 mm., or 0.21 atmosphere, for the partial pressure of the air. The mixture drawn off from the coil would then be 21 parts air and 79 parts water vapor. So, to vent the 56 cubic inches of free air, it would be necessary to vent 267 cubic inches, or enough to fill a 2-inch pipe for a distance of over 6 feet.

Again, the velocity of steam in the steam space is always appreciable, even from velocity of approach alone. This prevents perfect separation of the air, because turbulence in the steam space will keep the air more or less distributed, at any rate, it will usually prevent the air-steam mixture which is vented ever being as rich in air as the theoretical maximum richness. Further, in multiple effect operation, vapors taken from any body after the first contain air which was dissolved in the liquid being evaporated or evolved by chemical action; and any apparatus where the space occupied by the heating steam is less than atmospheric pressure will have some air drawn in through leaks.

How far such gases collect as a film on the surface of the condensed liquid film, or how far they are swept off by the steam velocity, cannot as yet be determined Since the coefficient of heat transfer from a permanent gas to metal is of the order of 1/1000 of that from steam to metal, it is obvious that the presence of non-condensable gases must lower the film coefficient considerably. And further, it must be recognized that this effect is not measured by the proportion of non-condensable gas in the fresh steam entering the tube, but by the partial pressure of residual gas in the tube, distributed more or less by turbulence and incomplete venting.

The importance of non-condensable gases in the condensation of steam was first pointed out by Osborne Reynolds in 1873.[1] None of the investigators who have worked on film coefficients have studied this matter. Orrok[2] (who worked on *overall* coefficients, not film coefficients) concluded first that the overall coefficient varied as $\left(\frac{P_s}{P_t}\right)^5$, where P_t is the total pressure in the steam space, and P_s is the partial pressure of the steam. This ratio $\frac{P_s}{P_t}$ may also be thought of as the per cent steam by volume in a steam-air mixture, and is usually called the "richness." Later,[3] he made actual determinations of the air present and decided that the overall coefficient varied as $\left(\frac{P_s}{P_t}\right)^2$. Kerr[4] decided that the overall coefficient varied as $\left(\frac{P_s}{P_t}\right)^3$.

Film Thickness.

The thickness of the liquid film depends on a number of factors. The steam, in passing, tends to scrape off this film. Since this is caused by precisely the mechanism which causes friction, it is measured by such a factor as $\frac{Dv\rho}{z}$, where the constants refer to the steam. But in addition the density and viscosity of the liquid also have a bearing, since the denser and more viscous the liquid, the less will be the effect of a given energy input from the stream of steam. In addition to these, drainage of the film by gravity must also affect its thickness. This will be affected by the shape, size, and disposition of the heating surface.

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[1] Collected Papers, *1*, 59-66.

[2] *Trans. Am Soc. Mech. Eng*, *32*, 1139-1200 (1910).

[3] *Trans Am. Soc. Mech. Eng*, *34*, 713-38 (1912).

[4] Bull. No. 149, La Agr. Exp. Sta. (1914); *Chem. Met. Eng.*, *11*, 336-7 (1913).

The rate of heat transfer through the film is probably purely by conduction. It is conceivable that the film of condensate might be thick enough to be flowing in turbulent motion. This could happen toward the bottom of long vertical tubes, or on the lower banks of large tubular heaters or condensers, where condensate from the upper parts of the heating surface drains over the lower parts. We usually think of the liquid film as in non-turbulent flow, and hence transferring heat only by conduction and not by convection.

The factors which affect the transfer of heat from condensing steam to a metal wall are, therefore, the shape factor of the steam space; the velocity, viscosity, and density of the steam; the amount of non-condensable gases in the steam; the viscosity and thermal conductivity of the condensate; and the physical shape and arrangement of the heating surface. Until these factors can be co-ordinated by some theory, even an approximate one, experimental work on this subject will necessarily be fragmentary.

Experimental Data.

Webster[5] gives data for his steam film, but makes no generalizations. He worked with steam at seven different pressures, ranging from 18 to 90 pounds absolute; a total of 96 experiments. He passed steam through the annular space between a cast iron casing 1.75 inches I.D., and a copper tube ⅞-inch O.D. At the exit of the experimental apparatus proper, the steam space was connected to a second condenser, so that more steam passed through the experimental apparatus than was there condensed. Knowing the weight of steam condensed in the apparatus, and the weight passing on into the secondary condenser, mean steam velocities in the apparatus proper can be calculated. Webster blew through his apparatus, to be condensed in his secondary condenser, from 50 to 80 per cent of the steam entering. This gives much higher velocities than would exist in ordinary practice. His velocities of approach (as discussed above) are of the order of 1 per cent of his linear velocities, and hence of little importance.

When his film coefficients for a given steam pressure are plotted against velocity, any one group gives a fairly good curve of the form $K_2 = av^x$. The different groups, however, do not show any regular relation to each other. In most of them the exponent of velocity is 0.4 to 0.5, but in two cases the exponent is 1.0. At a velocity of 50 feet per second, which falls fairly near the middle of the various groups, the film coefficient K_2 varies from 2400 to 2900; and is apparently independent of steam pressure.

[5] *Inst. Eng. Shipb. Scot., 57,* 58-79 (1913).

When $\frac{Dv\rho}{z}$ is calculated for Webster's steam film, using the values for ρ and z corresponding to the *liquid* film, the results show no regularity at all. There is no connection between his values for K_2 and the corresponding values for $\frac{Dv\rho}{z}$.

A. W. Porter[6] makes a mathematical analysis of Webster's data and concludes that

$$K_2 = \text{const.} \times \left[\frac{v_1^{3/4}\rho_2^{1/4}d^{-3/4}}{\mu_2^{1/2}\mu_1^{1/4}}\right] \tag{50}$$

where subscript 1 refers to the steam side and subscript 2 to the water side There is an interesting editorial comment on Porter's work[7] in which the limitations of Webster's work are pointed out and the dangers of too elaborate mathematical analysis of his results are illustrated.

Clement and Garland[8] calculated steam film coefficients for their experiments. At the end of their apparatus was a vent from which steam was blown continuously during the runs. Since this amount of steam was not determined, calculations of steam velocity are impossible. As the velocity of the water inside the tube increased, the water film coefficient increased. This condensed more steam and gave a higher steam velocity; hence steam film coefficients also rise as the water velocity rises. They range from 1600 to 2600 in 23 experiments, but further analysis is impossible.

Nusselt[9] was primarily concerned with gas film coefficients. His tube was heated by steam, but he gives no data for the steam side. He merely states that the average steam film coefficient was found to be about 1950.

McAdams and Frost[10] report seven experiments on steam, one on carbon tetrachloride vapor, and two on benzene vapor. They did not determine steam velocities. Their film coefficients range between 2060 and 3360 for steam, 280 for carbon tetrachloride vapor, and 300-360 for benzene vapor. If these film coefficients are multiplied by $\frac{z}{\lambda}$, they come much closer, but the steam film still varies from 2000 to 2600, the carbon tetrachloride is 2500, and benzene is 1350-1580. Hence this factor $\frac{z}{\lambda}$ does not bring different vapors close enough to permit generalizations, though it does show that the thermal conductivity and viscosity of the film of condensed liquid are of great importance.

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[6] *Engineering, 107,* 125-6, 140-2 (1919).
[7] *Engineering, 107,* 145 (1919).
[8] Bull. Univ. Ill. Eng. Exp. Sta, No 40 (1909).
[9] *Mitt. Forschungsarbeiten,* No 89 (1910).
[10] *J. Ind Eng. Chem, 14,* 13-18 (1922).

Heat Transfer between Solids and Boiling Liquids.

No instance of a determination of a film coefficient between solids and boiling liquids has come to the writer's attention. The theory is a little less complicated than that of heat transfer between condensing vapors and solids; but it is still sufficiently obscure so that no attempt has yet been made to formulate it. It will, therefore, be necessary to deal in the most general qualitative statements.

It is not clear whether or not there is a stagnant film of liquid through which heat must pass before there can be ebullition. In other words, do bubbles of vapor form on the *outside* of a stagnant film, or immediately on the metal? It is certain that improving the circulation of the liquid improves the rate of heat transfer. This may be due to the fact that the bubbles first formed adhere to the tube and partly insulate it. As the boiling liquid circulates, it may sweep off these bubbles, giving new portions of the liquid an opportunity to touch the surface and become heated.

It is also known that liquids, entirely free from dissolved gases, dust, or bubbles of their own vapor, may be heated many degrees above their boiling point without any formation of steam. The introduction of any of the vapor phase, or dust, or anything else which may act as a nucleus on which vapor bubbles could form, starts the boiling at once. It may be that the effect of circulation of the liquid is to bring into contact with a stagnant, non-boiling film, bubbles of steam to serve as nuclei. However this may be, the rate of heat transfer increases as the circulation of the liquid increases.

Whether the purpose of the circulation is to wipe off a stagnant non-boiling film, to wipe off bubbles, or to inoculate a stagnant non-boiling film with vapor, the result is a function of the number of impacts of liquid particles on unit area of the heating surface. This again should be a function of $\frac{Dv\rho}{z}$. Here we have the greatest difficulty in an attempt at numerical evaluation. If the liquid be outside of tubes, what is the shape factor D, and what is the velocity v in any case of boiling liquids? This latter is dependent mainly on the rate of heat transfer and on the physical arrangement of the apparatus. Therefore, for most cases in practice, it would be almost impossible in our present state of knowledge to evaluate this factor numerically to find whether or not it did have an influence. It is certain that we may say qualitatively that the rate of heat transfer will increase with the velocity of the liquid and decrease with its viscosity.

A less easily evaluated factor is the condition of the metal surface. Pridgeon and Badger[11] have shown that changes in the condition of the heating surface have a profound effect on the rate of heat transfer. The effect of scale or other deposits of foreign matter is so obvious that it needs no comment at this time. Most of the materials which might so deposit have thermal conductivities of the same order of magnitude as that of water. Remembering that the thickness of a stagnant layer of water, which will account for all the thermal resistance met in most cases, is in the order of hundredths or thousandths of an inch, it will be seen that a very thin layer of scale would be equivalent to increasing this stagnant layer many times. Whether the work here referred to showed the effect of complete removal of such foreign deposits, or whether there really is a difference between smooth and rough surfaces that are equally clean, has not yet been shown.

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[11] *Ind. Eng. Chem., 16,* 474-8 (1924).

In one sense this work sheds some light on the question discussed in the previous section. If the problem of boiling the liquid is conditioned by bringing to the liquid film nuclei on which steam bubbles might form, the smoother the tube the poorer the rate of heat transfer should be. This is because the points of metal on a rough tube could serve as nuclei for gas bubble formation. Since the cleaner and smoother the tube, the better the rate of heat transfer became, this would indicate that a supply of nuclei for bubble formation is not the determining factor.

The subjects of this chapter, as has been shown, have been practically untouched by experimental work or theoretical formulation. The *overall* coefficients between condensing steam and boiling liquids, which are an important consideration in this book, have been studied in certain aspects and will be discussed as such in a later chapter.

General Conclusions.

Evidently the subject of steam film coefficients is in a lamentable state. The best that one can do is to say that $K_2 = 2000$ to 3000, increasing with steam velocity, decreasing with the partial pressure of non-condensable gases, decreasing with viscosity of the condensate film, and increasing with the thermal conductivity of the condensate film. The film coefficients between metal and boiling liquids are as yet unknown, but increase with the rate of boiling and decrease as the viscosity of the liquid increases.

Summary.

In Chapter 3, we developed the equation

$$U = \frac{1}{\frac{1}{K_1} + \frac{d}{\lambda} + \frac{1}{K_2}} \qquad (26)$$

The term $\frac{d}{\lambda}$ is based on data which Chapter 1 shows to be fairly well in hand. Then we have discussed the values which $\frac{1}{K_1}$ or $\frac{1}{K_2}$ may have for the conditions of gas to metal, non-boiling liquid to metal, and steam to metal. Our knowledge of the first of these seems to be in fair shape within certain narrow limits, the second is very discouraging, and the third seems now nothing but a guess. In the fourth possible case, that of heat transfer from metal to boiling liquid, no numerical data, even the most fragmentary, have been published.

Considering the complexity of the phenomena of heat transfer, we can only hope to come to an understanding of them by analyzing heat transfer into its component film resistances. It cannot be too often repeated that in this way, and in this way only, will we be able to make real progress. The amount of work so far done, however, sheds so little light as merely to make the darkness visible, and there are relatively few cases of commercial design that can really be solved by our present knowledge of film coefficients. The case of the double pipe heat interchanger, with liquid both in the central pipe and in the annular space, should be fairly amenable to treatment along the above lines.

Chapter 7.

Heating and Heater Design.

There exist, here and there in the literature, isolated determinations and systematic studies of overall coefficients of heat transfer, many of them carefully done[1] The most conspicuous case is that of the surface condenser—low pressure steam around tubes transmitting heat to non-boiling water inside tubes. Because of the importance of the surface condenser in power generation, several general investigations have been made of heat transfer in them, and on the effect of some variables, notably water velocity, on such heat transfer coefficients. Without exception this work has concerned overall coefficients only—the film concept was unknown to these workers or neglected by them, though it was first discussed by Osborne Reynolds in 1874.

Heat Transfer Coefficients.

The most striking illustration of how little value this work has is Figure 14, taken from Royds,[2] in which the work of 12 such investigators is plotted on a common basis Evidently a general expression for the effect of water velocity alone is impossible; and this work has not been more "practical" or productive of directly useful results than the work on film coefficients. From Figure 14 may be drawn any kind of conclusion one pleases as to the rate of heat transfer in surface condensers.

In commercial practice, having regard for average conditions in actual operation, tubular heaters and surface condensers are often designed on the basis of $U = 200$ to 350. If scale or excessive corrosion is to be expected, a lower figure must be taken. If the surface may be kept clean and the steam space vented of air, and if high liquid velocities can be assured, a higher figure may be used. There seems to be a general tendency to assume that the overall coefficient varies as the square root of the mean linear velocity of the cooling water.

A very wide range of coefficients is possible according to conditions. The writer knows of cases of tubular heaters transferring heat from steam to salt brine where the velocity of the brine inside the tubes was only 0.2-0 5 foot per second, and the resultant value of U was 10 to 20. High velocity heaters, heating clear water with new clean copper tubes, may give U = 1000. In heating substances other than water, considerable changes may occur in U if the viscosity or the thermal conductivity of the new liquid is much different from that of water.

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[1] Much of such data is collected, examined, and compared in Royds' two books, "I. Heat Transfer by Radiation, Convection, and Conduction," and "II. Heat Transfer in Boilers, Condensers, and Evaporators," Van Nostrand, 1921.

[2] II, p. 176.

Even with our incomplete knowledge of film coefficients, an approximate calculation may be made for many cases which will be better than such guess work as above. The greatest uncertainty lies in the factor of safety to be applied to the calculated overall coefficients to cover scale, dirt, rust, possible over-loads, etc. The coefficient actually used may be from ¾ to ⅕ of the calculated coefficient according to conditions.

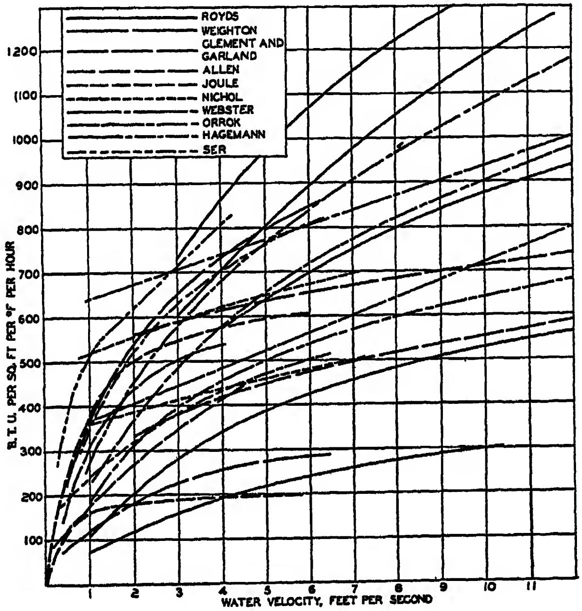

FIG. 14.—Overall heat transfer coefficients in surface condensers.

Problem.

What will be the coefficient of heat transfer when heating 11 per cent caustic soda solution with steam at 5 pounds gage, the solution flowing inside 1.25-inch iron tubing (13 gage) at a velocity of 5 feet per second? The solution is being heated from 150° to 200° F.

Steam Film Coefficient.

For this all that can be done is to assume $K_2 = 2000$.

Metal Wall.

$$13 \text{ gage} = 0.095'' \qquad \lambda = 35$$

$$\frac{d}{\lambda} = \frac{0.095}{12 \times 35} = 0.000226$$

Liquid Film Coefficient.

For this, equation (49) may be used. The mean steam temperature is 227°, the mean liquid temperature is 175°, and on the basis that the liquid film coefficient will be about half the steam film coefficient (the temperature drop through the water film about twice that through the steam film), the mean metal temperature would be 210°. Hence the mean liquid film temperature would be about 190°.

In spite of the general impression to the contrary, most of the physical properties of solutions needed for such problems are practically unknown. We will have to assume:

1. The thermal conductivity of caustic solutions is the same as that of water. This is obviously incorrect, but no data are available for a more accurate assumption.

2. The ratio of the viscosity of a caustic solution of a given strength to the viscosity of water at the same temperature, is constant at all temperatures. This also is incorrect but again no data exist for a better evaluation.

The data for substitution in equation (49) are, then,

$$D = 1.25 - (2 \times 0.095) = 1.06''$$

$$\lambda = 0.405 \text{ (at } 190^\circ\text{)}$$

The specific gravity of 11 per cent NaOH at 175° is 1.099 (Landolt-Bornstein), hence,

$$\rho = 62.4 \times 1.099 = 68.5 \text{ (at } 175^\circ\text{)}$$

The only data in Landolt-Börnstein for the viscosity of NaOH is a table showing the specific viscosity of solutions from 0.25 N. to 4 N. at 18°. From this, by interpolation, we find the specific viscosity of an 11 per cent solution at 18° to be 2.12 times that of water at the same temperature. Hence,

$$z = 0.325 \times 2.12 = 0.689 \text{ (at } 190^\circ\text{)}$$

Substituting in equation (49) gives

$$K_1 = \frac{23.3 \times 0.405}{1.06}\left(\frac{1.06 \times 5.0 \times 68.5}{0.689}\right)^{0.8} = 1340$$

Overall Coefficient.

The results of the problem in Chapter 6 showed that it is not necessary, in such a case as this, to consider the differences in the areas of the films. Equation (26) is sufficiently accurate.

$$U = \frac{1}{\frac{1}{2000} + 0.000226 + \frac{1}{1340}} = 680$$

It should be noticed that in this case the resistance of the metal wall is about 15 per cent of the total resistance.

Since steel in contact with caustic will stay quite clean, the only factor which might decrease the coefficient would be rust on the steam side of the tubes. The working coefficient may be assumed as 75 per cent of the calculated, or about 500.

HEATER DESIGN

From the calculation of the heat transfer coefficient the design of the heater does not follow directly. To the beginner many questions have no doubt already occurred; such as, Why 1¼-inch tubes? Why 5 feet per second velocity? This book is not primarily concerned with details of mechanical design, but in the following sections some of the more important features will be considered.

Tubes.

The metal of which the tubes are to be made is often dictated by considerations of resistance to corrosive action of the material to be heated. Such problems are obviously special and to be decided in each case on the basis of special information. Of the commoner metals the choice is usually between iron, copper, and brass.

Brass tubes are seldom used for heater and evaporator tubes because of the tendency to "season cracks." Large surface condensers are fitted with brass tubes, but the practice is not general for smaller units. Copper, while more expensive, is much stronger and more reliable. In ordinary heaters the difference between the cost of copper and brass is not a serious item.

Iron tubes are, in general, less usual because of the tendency to rust, and the greater difficulty of making coils and bends. Copper will keep a cleaner surface under many conditions, and the tube may be made thinner for the same factor of safety against corrosion. The increased thermal conductivity of copper is a small item. The thinner wall (hence lighter weight) and cleaner surface of the copper tube are its main advantages. In the problem above, iron was chosen because of its greater resistance to corrosion by caustic soda.

In this place it is not possible to go into the subject of corrosion, but it should be pointed out that a copper tube in an iron heater, in contact with a strong electrolyte, gives excellent opportunity for electrolysis. Such combinations of metals must be watched very carefully, and are to be employed only when the possibilities of electrolysis have been carefully studied and there is evidence that it will not be serious.

Tube Proportions.

When the metal has been selected, the next most important question is that of tube diameter. Other things being equal, equation (49) shows that heat transfer increases as $\frac{1}{D^{0.2}}$. Hence smaller tubes give a slightly better coefficient. A more important consideration is that the same heating surface may be gotten into a smaller tube sheet with smaller tubes. This decreases the size, and hence the weight and the cost of the heater. So, from these considerations, we wish a tube as small as possible. Also, with a given liquor velocity and a given coefficient, smaller tubes make it possible to get more heating surface in a given length of heater, and hence reduce the number of passes.

On the other hand, if the tubes are to be cleaned, the type of cleaner used fixes the minimum diameter. Tube cleaners will be discussed in Chapter 11, but it may be said here that if the turbine type of cleaner is to be used, the tubes may not be less than 1¼ inches, and preferably should be 1½ inches or over. If the cleaner be on a solid rod externally driven, the tubes may be as small as ¾ inch. As the tube diameter is smaller, the tube must be shorter; and, therefore, for a given heating surface, more tubes must be used. This also complicates design. If the tubes are to be expanded into the tube sheet, a tube ½ inch or less in diameter cannot be expanded with any satisfaction. General practice is to use tubes from ¾ inch to 1½ inches according to the service and the size of the heater.

Tube length is partly determined by liquor velocities and partly by mechanical strength. To keep up liquor velocities with short tubes means many passes, which in turn result in complications of design and loss of head. Commercial practice is generally to use a tube with a ratio of length to diameter not far from 100.

Tube wall thickness is determined largely by possible corrosion. Iron tubes must be thicker than copper because of possible rusting and pitting. Where corrosion is not a factor, the tube is, of course, as thin as possible consistent with mechanical strength. Very thin tubes cannot be secured to tube sheets satisfactorily, nor will they last long if mechanical cleaners are used often. Copper tubes are usually from 12 to 18 S.W.G. thick; but 18 gage tubes are very thin and not satisfactory. Iron tubes run a little thicker. Iron pipe is not used unless frequent replacements or much corrosion is expected. Pipe is thicker-walled than tubing; but is often easier to obtain for quick repairs, and is much cheaper than tubing.

Corrugated copper tubes are sometimes used where no scale, or only a soft mud, is expected. The corrugations may be annular or spiral. By increasing the turbulence of the liquid flow, they help the coefficient of heat transfer somewhat, and also tend to prevent the accumulation of soft scale. For moderate lengths, the corrugations may be depended on to take up differences in expansion between the shell and the tubes, eliminating the necessity of special expansion joints.[3]

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[3] Isaachsen, U. S. Pat. 1,535,531, 1925.

Special inserts are sometimes put inside heater tubes to increase the turbulence of the flow. These may be thin strips of metal twisted to a long-pitch spiral, or wire spirals.[4] Sometimes instead of twisted strips the whole length of the tube, a short strip of twisted metal is inserted at the inlet end of the tube. In any case, the idea is to start a whirling motion of the whole column of liquid, thus causing more scouring action on the stagnant film.

Tube Fastenings.

This is one point that is a source of much trouble to operators. A tight heater is obtained, however, not so much by one method of tube fastening or another, but by careful workmanship and maintenance, no matter what method is used. We may note the following methods of securing tubes to tube sheets or headers:

1. Rolling or expanding.
2. Packing.
   a. Packing plates.
   b. Ferrules.
3. Screwing.
4. Brazing.
5. Welding.

Rolling is probably the commonest method. Where the so-called hairpin tubes are used (U-shaped with the curved end free) both ends may be rolled into the tube sheet. If straight tubes are used and both ends rolled in, some provision must be made for expansion. Thin copper tubes are not easily made tight by this method. A ferrule about as long as the thickness of the tube sheet is sometimes placed inside the tube proper to give more metal at the point of expanding, but this expedient is of doubtful value. Rolling in tubes makes a tight heater if well done, but a poorly rolled tube, or too thin a tube rolled in, is a source of continual annoyance. Tube replacements are more difficult when tubes are so attached.

Packing the tubes makes removal easy, provides for expansion, and allows the use of thinner tubes. The maintenance of the packing may be an expense, especially if the liquid being heated affects the commoner types of packing. Since in tubular heaters the liquid is usually inside the tubes, the packing is exposed to the action of the liquid being heated. The caustic heater of the problem above could not be built with packed tubes, for there is no material that could be used for packing which would long stand the action of hot caustic soda.

Packing plates are made about as in Figure 15. One plate holds 4, 6, or 8 tubes. Unless the tubes are very small, more than four to a plate is not desirable, as it is too difficult to put enough pressure on the plate with one stud to keep more than four gaskets tight. The gaskets for such a plate must be specially moulded, which makes the expense prohibitive for small lots of odd sizes. They are regularly made for ⅞-inch and 1¼-inch tubes. Half-inch standard iron pipe may be held in a ⅞-inch gasket, and 1-inch pipe in a 1¼-inch gasket. Since the tubes have a tendency to crawl with repeated expansions and contractions, the packing plate has a shoulder that prevents the tube getting out of the tube sheet at either end.

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[4] Green, U. S. Pat. 874,673, 1907; Cartault, U. S. Pat. 1,024,436, 1912; Jones, U. S. Pat. 1,335,506, 1920; Hartburg, U. S. Pat. 1,459,024, 1923; Wright, Engl. Pat. 13,192, 1901; Price and Jones, Engl. Pat. 184,443, 1922. See also note 16, p. 89.

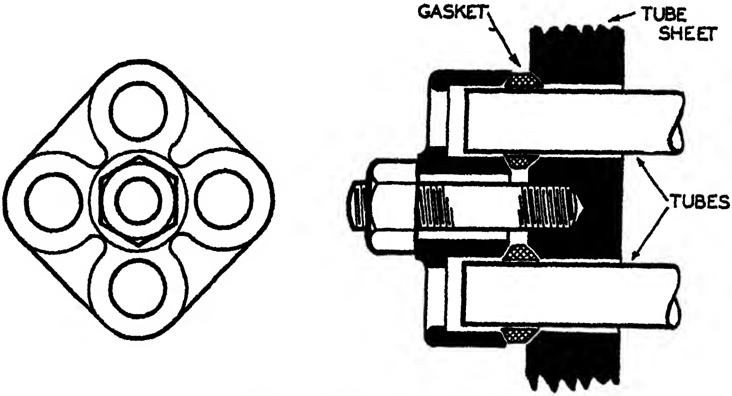

FIG 15—Packing plate.

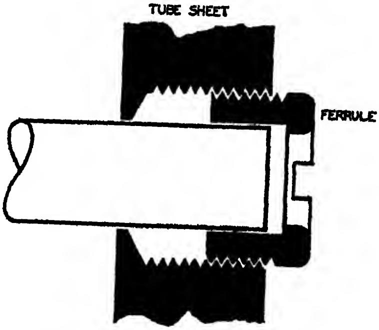

FIG 16—Packing ferrule.

The type of packing which is held in by a ferrule is shown in Figure 16. This takes less space in the tube sheet, but makes more labor in replacements. Usually when a tube is replaced the packing is destroyed, both in this type and the packing plate type. It is easier to remove the old gasket where a plate is used than to get the packing out of one of these recesses. This type of packing is used on surface condensers, where the ferrule is in contact with cold water only, and more attention is paid to maintenance than in heaters for process work.

Tubes are rarely screwed into tube sheets.[5] Only a heavy walled tube would have metal enough to provide for threading, and some means must be provided for attaching the other end to its tube sheet.

Where copper coils are connected to a header, they may be brazed on, but this construction is used only in special designs. It can hardly be considered a general method.

The use of torch welding for fabricating all kinds of equipment is growing in favor. Tubes welded to tube sheets give a combination which is absolutely tight under difficult service conditions, but tube renewals are somewhat more expensive. The caustic heater of the problem above could well be made with tubes welded to the tube sheets. Some provision for expansion has to be made in such cases.

Expansion Joints.

When a heater is put into or taken out of service, the shell and the tubes will not heat at the same rate even if of the same metal. For instance, if the shell is cast iron and the tubes are empty when steam is turned on, the tubes will heat very much faster than the shell. If the shell and tubes are of different metals, the difference in expansion may be greater. If the tubes are copper, 8 feet long, they will expand about 0.12 inch in being heated from 70° to 212° F. This is enough to start a rolled joint or bend the tubes, if both ends were fastened rigidly to supports which had not been heated (e.g., a heavy cast iron shell). One such strain might not be serious, but several repetitions of this would cause failure somewhere. Methods for taking care of this may be classified as

1. Packed tubes.
2. Hairpin tubes or coils.
3. Field tubes.
4. Floating heads.
5. Bumped tube sheets or shells.

Packed tubes have been discussed above and their action in taking care of expansion is obvious. *Hairpin tubes* are simply U-shaped tubes with both ends secured to the same tube sheet. Obviously there must be a division in the liquor inlet head to send the liquor into one end and out at the other of the tubes. They must be made of heavy enough metal, and have long enough bearing in the tube sheet, to be self-supporting. Special devices (such as supports with rollers) are sometimes added so that when the tube sheet is detached from the heater, the whole bundle of tubes may be easily drawn out for cleaning.[6] Others use a tube rolled up like a spiral spring,[7] with the ends left straight for attaching to the tube sheets. Still others use the corrugated tubes described above

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[5] Terriault, U. S. Pat. 1,500,286, 1924; 1,532,520, 1925.

Field tubes are tubes closed at one end and supported from the open end only Obviously there must be some provision for circulating the fluid inside the tube This consists in a separate set of smaller tubes, open at both ends, and secured to a separate tube sheet, Figure 17. The liquid would enter connection A in one chamber and leave at B in the other. To make this a multipass heater, partitions may be placed in both chambers, so that in one pass the liquid enters the annular space and leaves by the central tube, while in the next pass it enters the central tubes. The tubes must be heavy-walled or short, and their bearing in the tube sheet long. The construction is rather complicated and not often met[8] This construction is sometimes called a "porcupine tube."

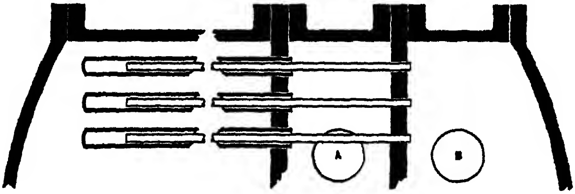

Fig 17—Field tube heater

Floating Heads.

A floating-head is a common method of taking care of expansion. This device amounts to making one tube sheet independent of the shell and movable inside it. Some forms are shown in Figures 18 to 23. Figure 18 is obviously the simplest. Figure 19 makes the main part of the shell smaller (and hence the heater is cheaper). In both these forms the floating head may simply rest loosely on the shell, or it may have fins cast on it to act as guides and assist in maintaining its alignment. Obviously such heaters must have an even number of passes, as the liquid must enter and leave through the fixed head. If they have more than two passes, there may be division plates in the floating head as in Figure 20, or the tube sheet may be in two or more parts, as in Figure 21. Figure 22 is a form involving a flexible connection at A, in which the number of passes may be odd. Figure 23 is another form in which the tube sheet A is prolonged into a head thick enough so that its edge never passes the packing, which is held in place by

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[6] Green, U S Pat. 874,673, 1907, Zimmerman, U. S. Pat. 1,289,350, 1918; Hildebrandt, U. S. Pat 1,478,855, 1923; Dillon, U. S Pat. 1,485,601, 1924.

[7] Lundberg, U. S Pat. 1,215,041, 1917

[8] Fraser, U S Pat. 779,122, 1905, Dixon, U. S. Pat. 1,520,938, 1924, Paul, U. S. Pat 1,508,471, 1924, Thompson, *et al*, Engl. Pat. 123,369, 1918.

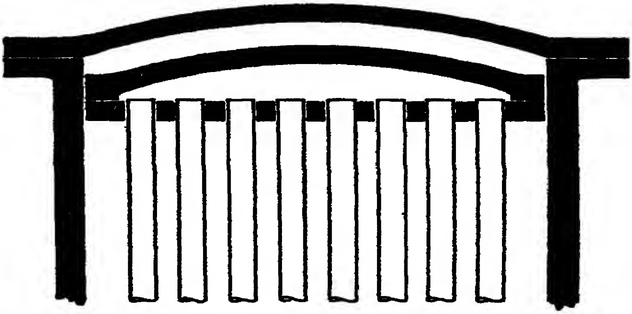

FIG. 18.—Simple floating head heater.

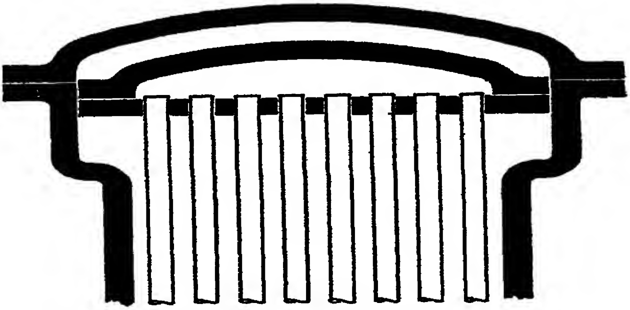

FIG. 19.—Floating head heater.

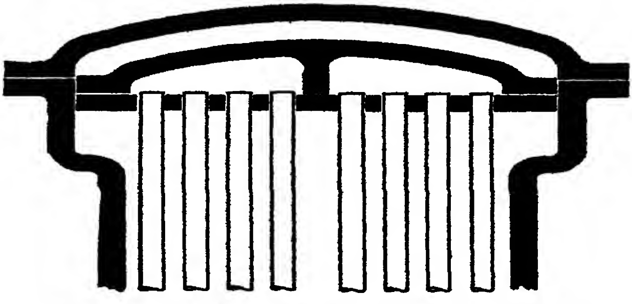

FIG. 20.—Multipass floating head.

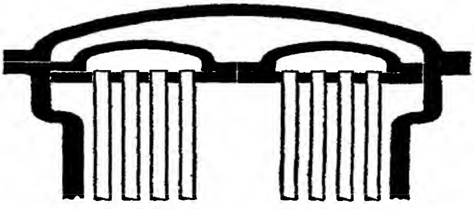

FIG. 21.—Divided multipass floating head.

the gland B.[9] One serious disadvantage of floating head heaters is the difficulty of testing for leaks where the tubes are fastened into the floating head.

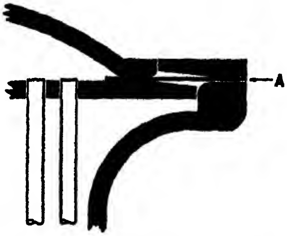

FIG. 22.—Flexible joint floating head.

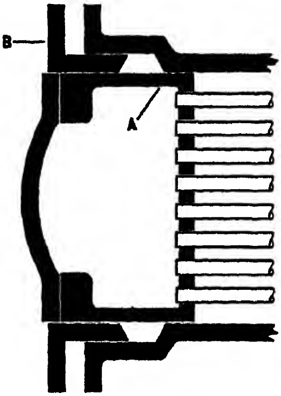

FIG. 23.—Packed floating head

Flexible Joints.

In very large heaters either the tube sheet, or the shell, or both, may be allowed to give but be fastened rigidly at the joints. Figure 24 shows a sheet steel tube sheet, flanged with a fairly large radius at R, which may be depended on for small amounts of expansion. A similar idea may be incorporated in the shell by making a special joint, as shown in Figure 25. This is not recommended where much expansion is to be expected. In small heaters a corrugation rolled in the shell, as in Figure 26, may be sufficient, especially if the shell is very light.

Tube Spacing.

The closeness with which tubes can be packed in a tube sheet determines the size of the heater. It is usually determined by the minimum

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[9] Modifications of this are shown by Muhleisen, U. S. Pat. 1,511,836, 1924; Sandberg, Engl. Pat. 15,890, 1911.

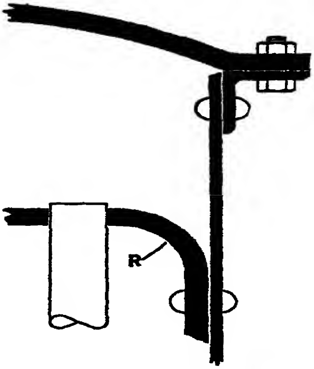

FIG. 24.—Expansion joint for plate steel heaters.

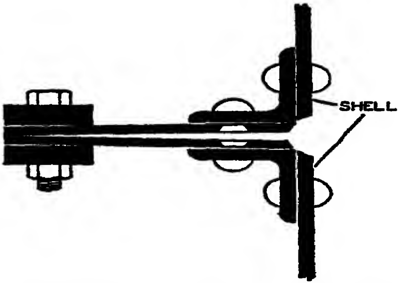

FIG. 25.—Expansion joint for heater shells.

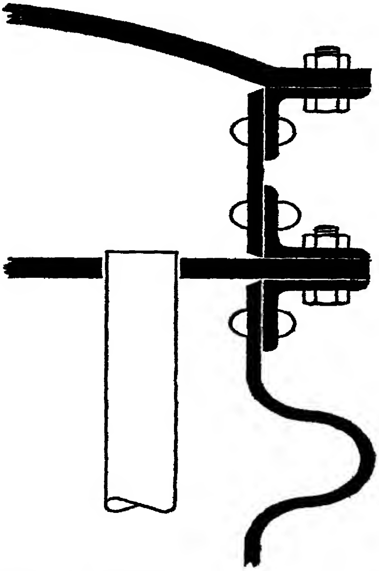

FIG. 26.—Rolled expansion joint for heater shells.

amount of metal which must be left between adjacent tubes to give the necessary strength to a tube sheet. It will, therefore, vary with the metal of which the tube sheet is made, the method of fastening the tubes, the thickness of the tube sheet, and the difference in pressure between the two sides of the tube sheet. The "bridge" or smallest amount of metal left between two adjacent tubes should never be less than ¼ inch, and ⅜ inch is better. In large tube sheets or with large pressure differences, it may be as much as 1 inch Cast iron tube sheets naturally should have a greater bridge than plate steel, unless the cast iron is very thick in proportion to its greatest unstayed width. The tubes should not be considered to act as stays. Tubes packed with ferrules naturally call for about the same bridge between adjacent stuffing boxes as rolled tubes; hence, the tube centers will be farther apart than rolled tubes of the same size by twice the width of the packing space. Tubes fastened with packing plates take the widest spacing. One company, when using packing plates, spaces ⅞-inch tubes on 1⅝-inch centers both ways (center lines at 90° to each other) and 1¼-inch tubes alternately 2-inches and 2½-inches in the horizontal rows, and 2¼-inches in the vertical rows.

Liquor Tubes vs. Steam Tubes.

Since, in most cases, the steam film coefficient is much higher than the liquid film coefficient, greater attention should be paid to the latter. In those cases where the steam is inside the tubes and the liquor surrounds them, the liquor will always have a low velocity and poor velocity distribution. This latter is important in cases where the liquor carries material which may deposit in areas of low velocity, or in cases where the liquor should be exposed to high temperatures for the minimum length of time. In many cases, the liquor would be flowing in viscous or straight-line flow, which obviously reduces the rate of heat transfer to that of pure conduction through the liquid. Although steam velocity has an effect on the rate of heat transfer on the steam side, at even the lowest steam velocities the steam film coefficient is in no great need of improvement as compared with liquor velocities. Consequently the liquor is almost invariably inside the tubes, where its velocity may be positive and high. In a few cases the steam may be inside the tubes, but these are always special. For instance, in condensing the vapors of a substance which is corrosive to ordinary materials, it may be cheaper to make the tubes, tube sheets, and heads of the special metal demanded by the circumstances. If the corrosive vapors were outside the tubes, not only these parts but the shell also would have to be of corrosion resisting metal.

Another case where steam tubes may be used is the so-called *"storage heaters,"* Figure 27. Where the demand for hot water is variable, instead of installing a heater to meet the peak load, one may install heating surface enough to meet the average load and follow this with a storage tank for hot water. If the heater is combined with the storage tank itself, the coils or tubes usually have steam on the inside. In such cases the steam to the coil is usually controlled by a thermostatic valve whose sensitive element is in the storage space. In such cases the liquid velocity is due principally to natural convection, and hence the heat transfer coefficient will be low.

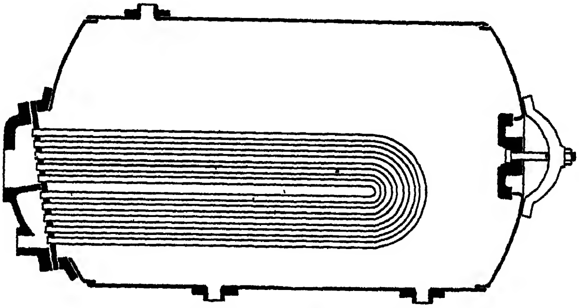

FIG. 27.—Storage heater.

Liquor Distribution.

The first question to be decided here is the liquor velocity desired. This, though of primary importance, will be discussed later. Assuming that both liquor velocity and tube proportions have been fixed, how are the tubes to be arranged in the heater? Altogether too many heaters are designed by assuming a coefficient arbitrarily, calculating the total heating surface needed, deciding on the proportions of the tubes, and then putting enough tubes in the heater to give this heating surface with the addition of anywhere from 10 to 200 per cent for safety. The liquor passes through all the tubes in parallel. The writer has seen a heater containing 350 2½-inch tubes in parallel, the liquor inlet being an 8-inch pipe. The area of the 350 tubes is approximately 30 times that of the inlet pipe, and hence the liquor velocity could not have been over 0.1-0.2 foot per second. The coefficient of heat transfer was almost negligible.

The obvious way to arrange heater tubes, in the light of our present knowledge, is to determine how many in parallel are needed to give the desired velocity, and then make them long enough to give the desired heating surface. The factor of safety can be incorporated into the design, either by using a lower coefficient than the calculated one, or by increasing the *length* of the tubes over the calculated length. Since in many cases this calls for a tube too long to be convenient, the result is that tubes of a convenient length are so assembled that the liquid goes through them in several passes.

Problem.

In the caustic heater at the beginning of this chapter, suppose that 100,000 pounds solution are to be heated per hour, and that the tubes are to be 10 feet long (ratio of length to diameter approximately 100). How many tubes, in how many passes, are to be used?

Number of Tubes per Pass.

The density of this solution is 68 5 pounds per cubic foot at mean fluid temperature. Hence the volume heated per second is $\frac{100,000}{68.5 \times 3600}$ cubic feet, and the area of path necessary to give a velocity of 5 feet per second is $\frac{100,000}{68.5 \times 3600 \times 5} = 0.0811$ square foot $= 11.7$ square inches. The inside area of a 1¼-inch 13-gage iron tube is 0.882 square inch. Hence $\frac{11.7}{0.882}$ or 13 + tubes in parallel are necessary. If we make each pass contain 13 or 14 tubes, according to the way the tube sheet works out, we will maintain the desired velocity.

Total Surface Needed.

The specific heat of 11 per cent NaOH solution is 0 91 (Landolt-Börnstein). The total heat to be transmitted per hour is then $100,000 \times 0\,91 \times 50$ B.t.u.

Steam at 5 pounds gage has a temperature of 227°. For calculating Θ_m we have (see Table IV, Chapter 1).

$\Theta_a = (227 - 150) = 77$; $\Theta_e = (227 - 200) = 27$; $\frac{\Theta_e}{\Theta_a} = 0.351$; factor $= 0.620$; $\Theta_m = 77 \times 0.620 = 47.7°$. (The arithmetic mean would be 52°, or an error of 8 per cent.)

The heating surface needed is then $\frac{100,000 \times 0.91 \times 50}{47.7 \times 500} = 189$ square feet.

Number of Passes.

The mean tube diameter is 1.155 inches. The heating surface in one pass of 14 tubes, each 10 feet long, is

$$\frac{1.55 \times \pi \times 14 \times 10}{12} \text{ or } 42.3 \text{ square feet.}$$

The number of passes needed is $\frac{189}{42.3}$ or 4.4. If the heater were made a 4-pass heater, the margin of 25 per cent allowed in the coefficient might be enough to cover the deficiency in heating surface, or the tubes might be made 11 feet long instead of 10. If the heater were 5-pass, the tube sheet layout would not be so simple, but the margin of safety would be greater.

Multipass Tube-sheet Layouts.

Having determined the number of passes and the number of tubes per pass, the tube sheet layout is primarily the concern of the draftsman. For a small number of passes, the question of whether or not the tubes are on center lines at 90° or at 60° to each other is not

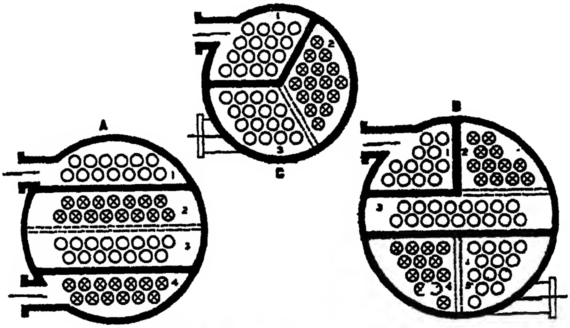

Fig. 28.—Multipass tube sheet layouts.

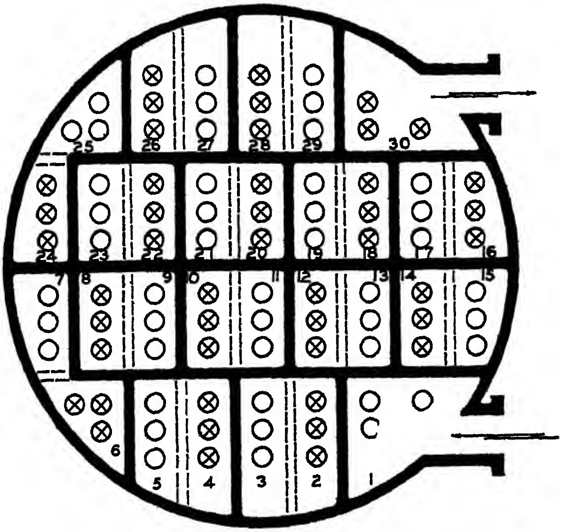

Fig. 29.—Multipass tube sheet—three tubes per pass.

specially important. The shell may usually be smaller with 60° center lines; and if the number of passes is some multiple of three, radial baffles make the layout very simple. For a small number of passes, a simple division into parallel zones is often satisfactory. This heater could be made a 4-pass heater with 13, 15, 15, and 13 tubes per pass, as shown in Figure 28A; or a 5-pass heater with 12, 12, 17, 13, and

13 tubes per pass as in Figure 28B. A 3-pass heater with 16 tubes per pass is shown in Figure 28C[10] This would reduce the velocity to about 4 feet per second and cut down the coefficient somewhat, but the tubes could be made slightly longer to compensate for this, and the construction would be decidedly cheaper.

Where the number of tubes per pass is small, 90° spacing of the tubes is often simpler Sample layouts for 3 and 5 tubes per pass are shown in Figures 29 and 30. A sample layout for 12 passes using 4-hole packing plates is shown in Figure 31 These are all merely suggestive; in each case that rises some ingenuity will be required to get the simplest layout in the smallest shell.[11] When the number of tubes per pass becomes 1, the double pipe heat interchanger construction becomes the simplest.

Steam Distribution.

There is rarely any attempt to distribute steam or increase its velocity. This may be done by baffles in the steam space, but is not usually done in small heaters Such baffles, if used, must be pitched so as to drain off condensate that may collect on them. Since Webster's work shows that the steam film coefficient increases about as the square root of the velocity, it would seem to be of advantage to use such baffles They are mentioned later in connection with heat interchangers on page 88.

Closely connected with steam distribution is the question of vents for non-condensable gases. In locating these it must be remembered that the air will collect at the points of lowest velocity, in pockets and corners. If the steam is given a definite velocity in any particular direction, air will be swept along and collect at the end of the path. This is especially true when there are baffles in the steam space. The larger the steam space, and the lower and less definite the steam velocity, the greater the area of the air vents should be.

Discussions often arise as to whether or not the gas in question is lighter or heavier than steam, the idea being that the air vents should be placed above or below in the steam space as the gas is lighter or heavier than steam If the steam velocity is very low, it may be possible for the non-condensed gas to collect in a layer, top or bottom, determined by its density. In most cases, there is sufficient turbulence in the flow to prevent such stratification. Ordinarily, only in the case of definite and positive steam flow along a definite path will the non-condensed gas collect in one place.

The air vents, therefore, should be as far from the steam inlet as possible. Since air will be swept ahead of the steam, the preferred

[10] In these and the following illustrations, open circles represent tubes in which the liquid is flowing *away* from the reader, crossed circles, those tubes in which it flows *toward* the reader The numbers show the order of the passes

[11] This subject is discussed, and three sample layouts given by Coxon, *Int. Sug. J.*, *19*, 216-9 (1917).

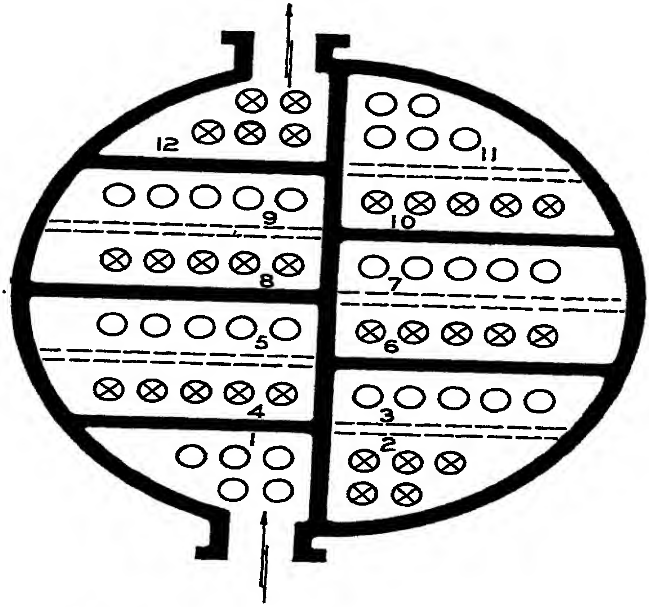
12
11
9
10
8
7
5
6
4
3
1
2

FIG. 30.—Multipass tube sheet—five tubes per pass.

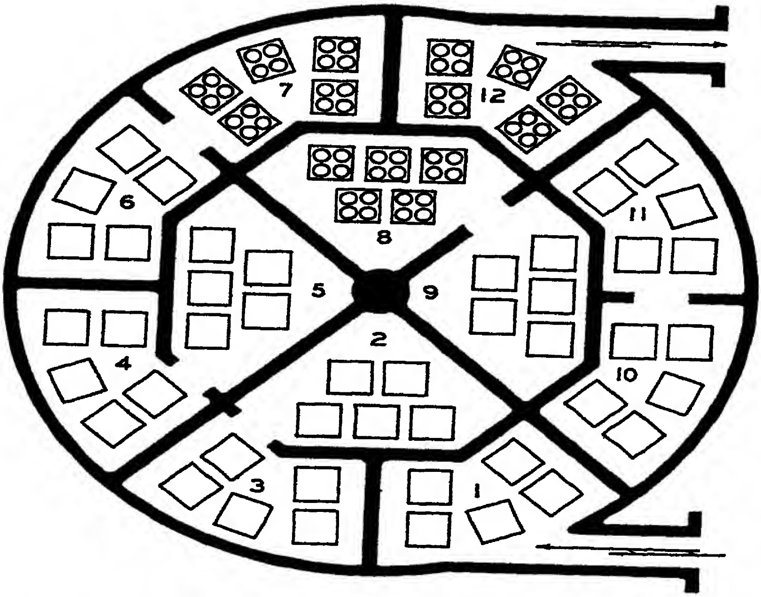
7
12
6
8
11
5
9
2
4
10
3
1

FIG. 31.—Multipass tube sheet for packing plate heater.

place will be near or at the condensate outlet. If the steam is at less than atmospheric pressure, this may be brought about by using an over-size condensate pump and allowing it to pump out air as well as condensate. If the steam is above atmospheric pressure, a vent on the condensate trap is very common. At the same time, due to turbulence, there may be air in the top of the heater and there should also be a vent there. The whole subject of the presence of air in steam, its effect on the steam film coefficient, and the effect of steam velocities on its segregation, has been very much neglected.

Heat Interchangers.

This term is here used to designate those cases where heat is transferred from a hot liquid to a cold one, as distinguished from the discussion above, where the source of heat was steam. In the case of heat interchangers, the liquid outside the tubes must be given some definite velocity, else it would move in viscous flow or short-circuit part of the heating surface.

In the case of tubular heaters of the general construction described above, various types of baffles may be used. The Griscom-Russell Company in their "Multiwhirl" coolers[12] place a spiral baffle outside the tubes so that the second liquid flows in a short-pitch spiral. This makes its path nearly perpendicular to the tubes and hence of the maximum turbulence. Many other patents show constructions carrying out the same idea.[13] Instead of forcing the second fluid to cross the tubes at right angles, many types use baffles parallel to the tubes, so that the second fluid traverses the heater shell from end to end several times.[14] Such baffles are often also used in steam heaters.

Double Pipe Heat Interchangers.

Instead of a bundle of tubes in a cylindrical shell, we may use two concentric pipes as a heat interchanger, one liquid in the smaller one and the other liquid in the annular space. These were mentioned in Chapter 5, and a common form of connection was shown in Figure 13. A number of variations have been mentioned,[15] but the standard construction is quite satisfactory. To increase the turbulence, a spiral such as mentioned on page 75 may be put in the inner tube, or a spiral baffle or coiled wire in the annular space, or the surfaces may have spiral corrugations rolled in or screw threads cut in them,[16] or the inner tube may be nearly filled with solid rod to decrease its cross-section.[17] Moore[18] states that while spiral corrugations increase the rate of heat transfer at low liquid velocities, at high velocities such a tube gives no higher coefficients than a plain tube.

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[12] Jones, U. S. Pat 1,335,506, 1920; 1,454,053, 1923; Sward, U. S. Pat. 1,524,595, 1925; Colston, U. S. Pat. 1,522,866, 1925, Jones, Engl. Pat. 176,753, 1921.

[13] Zimmerman, U. S. Pats. 1,168,368, 1916; 1,338,479, 1920; Forseiller, U. S. Pat. 1,410,561, 1922; Schoeckel, U. S. Pat. 1,453,250, 1923; Steffa, U. S. Pat. 1,453,292, 1923, Sims, U. S. Pat. 1,469,193, 1923; Ripley, U. S. Pat. 1,505,429, 1924; Sward, Engl. Pat. 204,025, 1923.

[14] Zimmerman, U. S. Pat. 1,289,350, 1918; Dickey, U. S. Pat. 1,489,932, 1924.

[15] Schneible, U. S. Pat. 826,966, 1906; Greisser, U. S. Pats. 856,140, 1906; 872,175, 1907; 912,671, 1909; Schneider, U. S. Pat. 1,016,582, 1912; Vollmann, U. S. Pat. 1,033,132, 1912; Dalzell and Feldmeier, U. S. Pat. 1,067,505, 1913; Vogt, U. S. Pat. 1,197,976, 1916; Reavell, U. S. Pat. 1,317,390, 1919; Cumfer, U S. Pat 1,481,255, 1924; Goosman, U. S. Pat. 1,512,219, 1924; Achard, Engl. Pat. 112,263, 1917, Rafn, Engl. Pat. 121,117, 1918; Stassano, Engl. Pat. 212,273, 1924.

Such heat interchangers are usually built of straight pipe arranged in vertical banks. Richter[19] reports tests of double tube heat interchangers made of straight pipe, and of pipe coils. When pipe of the same length and diameter was tested, first as straight runs, and then when wound into a spiral 3 feet in diameter, the coil showed 20 per cent higher rates of heat transfer than the straight pipe for the same liquid velocities. Richter also altered the radius of the coil, but at the same time changed the tube proportions, so that the results are not strictly comparable. In general, the rate of heat transfer seemed to increase as the coil diameter decreased. This is obviously due to the increased turbulence as the liquid flowed in shorter radius spirals.

In order to decrease space, double pipe interchangers may be built with several small pipes inside one large one. Here one must be careful lest the cross-section left outside the small pipes be so large as to decrease unduly the velocity of the second liquid. The connections at the ends are somewhat more complicated[20] in this case.

Special Constructions.

An endless variety of heater construction, not specifically listed above, may be found. Some are based on structural considerations, some are developed to compete with a similar heater, some are purely imaginary, and a few are based on a real understanding of heat transfer. Most of these latter take elaborate and complicated means to secure added turbulence of the fluids. One may find spiral coils,[21] flat coils,[22] combinations of coils of different diameters, tubes bent into special zig-zag shapes,[23] rotating cylinders with or without scrapers to

[16] Frost, U. S. Pat. 1,531,777, 1925; Kershaw, Engl. Pat. 15,455, 1900; Bennett, Engl. Pat. 6,146, 1901; Lovekin, Engl. Pat. 5,145, 1910; J. Baker Sons, *et al.*, Engl. Pat. 13,602, 1914; Stables, *et al.*, Engl. Pat. 114,651, 1917; Thompson, *et al.*, Engl. Pat. 123,369, 1918; J. Baker Sons, *et al.*, Engl. Pat. 176,469, 1920.

[17] Daniel, *et al.*, Engl. Pat. 19,124, 1905; Sawyer, *et al.*, Engl. Pat. 105,127, 1916.

[18] *Trans. Am. Inst. Chem. Eng.*, *12* (II), 180 (1920).

[19] *Ibid.*, 147-85.

[20] Thau, *Chem. Met. Eng.*, *29*, 98-102 (1923); Davis, U. S. Pat. 1,272,232, 1918.

[21] Anon., *Chem. Met. Eng.*, *29*, 194 (1923); Bowman, U. S. Pat. 843,791, 1907; Woodcock, U. S. Pat. 854,976, 1907; Price, U. S. Pat. 1,382,670, 1921; Musselman, U. S. Pat. 1,469,805, 1923; Goosman, U. S. Pat. 1,464,705, 1923; Wilson, U. S. Pat. 1,504,732, 1924; Chambers, *et al.*, Engl. Pat. 5,251, 1911.

[22] Davis, U. S. Pat. 1,498,954, 1924.

[23] Shaw, U. S. Pat. 1,456,255, 1923.

agitate the liquid film,[24] special baffles to induce added turbulence,[25] and many others.

Liquor Velocities.

In designing a heater for a given set of conditions, one must usually decide first on the liquor velocity. Many thousands of heaters have been built without any conception of the effect of this factor on heat transfer. Within the last fifteen or twenty years, the sugar industry, both beet and cane, has revised its standard practice so that heaters, instead of being built with juice velocities of less than 1 foot per second, are now designed for 4 to 6 or even 10 feet per second.[26] Coxon [27] quotes the following capacities from cane sugar practice:

	Tubes per Pass	No Passes	Linear Velocity	Gals Juice Heated per Hour
High velocity	2	56-60	7 16	3000-4000
Medium velocity	14	12-20	1 6	2500-3000
Low velocity	60	3-6	0 13	1500-1700

Many such comparisons may be found in the sugar literature, but the information is in such shape that it does not help the designer who wishes to know, "How fast can I afford to pump the liquid?" Obviously increased velocity means increased friction, and this means increased cost of operation. The economical point is plainly at that velocity where increased cost of pumping more than offsets decreased first cost. In certain cases this may be calculated.

The cost of the heater in the problem of this chapter could be approximated by assuming a first cost of $3.00 per square foot heating surface, a charge for interest and depreciation of 25 per cent, and operation 24 hours a day, 300 days per year. The total heating surface of the 4-pass heater with 11-foot tubes is

$$\frac{1.15 \times \pi}{12} \times 56 \times 11 = 202 \text{ square feet.}$$

The overhead per day would be

$$\frac{202 \times 3 \times 0.25}{300} = \$0.505$$

The cost of pumping is more uncertain. The data in Appendix I permit an approximate calculation of friction, and hence the power required; but losses in the heads are rather questionable. It will be necessary to assume that the velocity of the liquid in the distributing boxes in the heads is so low that there will be no friction there, and the entrance and exit losses will be figured accordingly.

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[24] Schneible, U. S. Pat. 1,478,088, 1924; Lantz, U. S. Pat. 1,487,248, 1924.

[25] Shaw, U. S. Pat. 1,499,481, 1924.

[26] Herman, *Z. Ver. deut Zuckerind*, *53*, 1351-5 (1903); Kerr, Bull. La. Agr. Expt. Sta. No. 149, 148-55 (1914); *Int. Sug. J.*, *17*, 515-21 (1915); Frazier, *La. Planter*, *62*, 29-30 (1919), Anon., *Int. Sug. J.*, *24*, 81-2 (1922); Anon., *Int. Sug. J*, *24*, 264-5 (1922).

[27] *Int. Sug J.*, *19*, 216-19 (1917).

To find the factor f we first calculate $\frac{dvs}{z}$, which is 8.45 for a 5-foot velocity. This gives, from the chart, $f = 0.0058$. In the formula for friction in straight pipe, $L = 4 \times 11 = 44$; $d = \frac{1.06}{12} = 0.0883$; and $g = 32.2$. We then have

$$\Delta p = \frac{2 \times 0.0058 \times 44 \times 68.5 \times 5 \times 5}{32.2 \times 0.0883} = 307 \text{ pounds per square foot.}$$

Where the velocity is increased from zero to 5 feet per second at the entrance of each pass, the constant in formula (113) will be 0.5, and the loss in head will be (for four passes)

$$\Delta h = \frac{0.5 \times 5 \times 5 \times 4}{2 \times 32.2} = 0.776 \text{ feet.}$$

Similarly, the exit loss in each pass will be due to a sudden drop from 5 feet per second to zero, or

$$\Delta h = \frac{(5-0)^2 \times 4}{2 \times 32.2} = 1.55 \text{ feet.}$$

The total head lost in these two items is $1.55 + 0.776$ or 2.326 feet. Since this is a liquid weighing 68.5 pounds per cubic foot, the loss in pressure will be $2.326 \times 68.5 = 159$ pounds per square foot, or over half the friction loss.

The power necessary to overcome this will be

$$\frac{100{,}000 \times (307 + 159)}{60 \times 68.5 \times 33{,}000} = 0.344 \text{ horse power (theoretical).}$$

If we assume a motor efficiency of 95 per cent, a pump efficiency of 60 per cent, and a current cost of $0.015 per kw. hour, the daily cost for power will be

$$\frac{0.344 \times 0.746 \times 1.5 \times 24}{0.60 \times 0.95} = \$0.162 \text{ per day.}$$

The total cost of this heater per day is then $0.505 + $0.162 or $0.667.

Next, a number of different velocities should be assumed, the coefficient calculated for each, and from this the size and number of passes needed for each case. Then, by calculating friction losses and cost for each case as above, curves will be obtained which are shown in Figure 32. From this it will be seen that the optimum velocity for this particular problem is 4 feet per second, but that there is very little change between 3 and 6 feet. Hence, probably the 3-pass heater of Figure 28C with a velocity of about 4 feet per second, would be desirable.

It is interesting to note that the cost of power increases almost as the square of the velocity, while the heater cost goes up very rapidly at low velocities and is not greatly changed at high velocities. This is partly because of the resistance of the thick iron tube used, which becomes more important as K_2 is greater. It should also be noted that this is *not* a general solution, any change in the cost per unit of heating surface, the per cent overhead, the hours operation per day, the days operation per year, the cost of power, or the pump efficiency will change the position of the minimum in the total cost curve. In general, an increase in operating cost shifts the minimum to lower velocities. An increase in overhead costs shifts it to higher velocities.

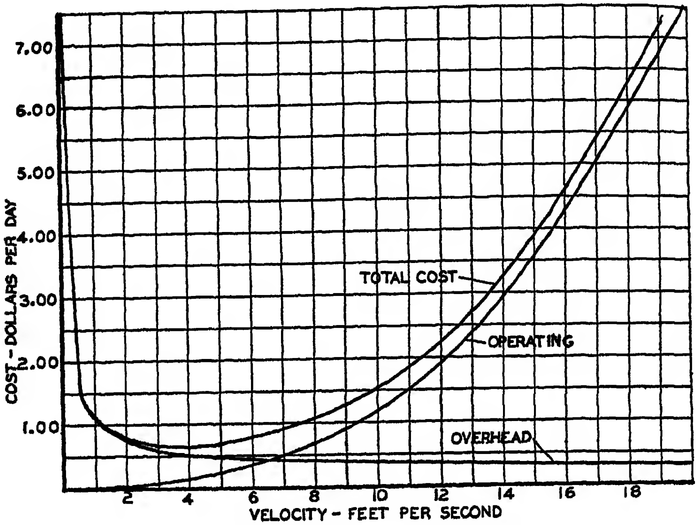

FIG. 32—Cost of heater operation.

Chapter 8.

Evaporator Bodies.

This chapter will deal with the physical construction of equipment for boiling liquids with steam. A few of the most important types will be described and discussed from the standpoint of construction only. Further modifications of these types, and various special designs, have arisen out of considerations of heat transfer, and hence will be covered in Chapter 9. This chapter will discuss only enough constructions to give a basis for the next chapter.

History of Evaporators.

The origin of the use of steam for heat is lost in antiquity. In the Middle Ages the water-bath was supposed to have been discovered by Maria the Jewess, an alchemist of the first century A.D. Von Lippman[1] points out that this is due to a mistaken connection between her and the expression "bain Marie," and that the water-bath was actually known to Hippocrates centuries before.

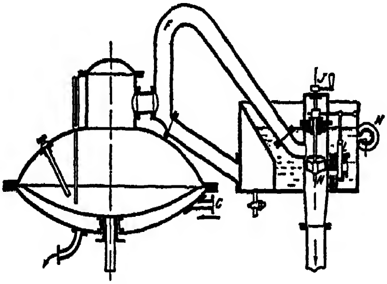

Fig. 33.—Howard's vacuum pan.

Steam-heated open pans of various designs were in use in the sugar industry about 1800. The first real advance was the invention of the vacuum pan by Howard, who took out patents in 1812 and 1813.[2] His pan is shown in Figure 33. He used a jet condenser and an air pump, and reached a boiling point of 60° C. His pan was immediately adopted in English refineries, but found very slow acceptance on the Continent, apparently because of a general disinclination to use air pumps. Degrande, in 1824, improved Howard's pan by adding a heating coil, thus increasing its capacity. Roth's pan, which used no air pump but depended on large volumes of cooling water flowing into a large receiver,[3] was more popular, but less efficient.

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[1] "Enstehung und Ausbreitung der Alchemie," Berlin, 1919, p. 50.
[2] Von Lippman, *Z. Ver. deut. Zuckerind.*, *62*, 967-79 (1912).

The origin of the idea of multiple effect evaporation is very difficult to trace. Péclet[4] says: "Under certain circumstances we may make use of a part of the latent heat of the vapor which is disengaged, to heat liquids contained in other boilers, and produce in them a gentle evaporation. It is this arrangement that I designate by the name of an apparatus of double effect (appareil à double effet).

"We may likewise employ the latent heat of the second vapors for heating other vessels, but the second boilers must then of necessity be closed, and the liquid must attain in them the temperature of ebullition. Hence the liquid in the first boilers must be subjected to a pressure greater than that of the atmosphere"

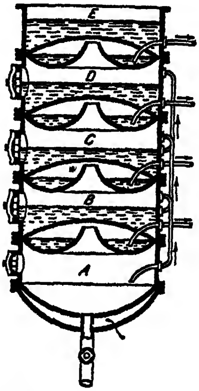

FIG 34—Pecquer's multiple effect

This is plainly the principle of multiple effect evaporation, but Péclet did not perceive the effect of putting the later bodies under a vacuum. So far as can be determined, Péclet never carried out the idea in practice.

In 1830 Norbert Rillieux, a native of New Orleans, was living in Paris and conceived the idea of multiple effect evaporation under vacuum. Horsin-Deon, whose father was Rillieux's secretary later, says,[5] that Rillieux tried to get Pecquer to adopt his ideas, but without success. In 1834, Pecquer took out a patent on a multiple effect under pressure, which is by some considered to give him priority. His apparatus is shown in Figure 34. It was never practical, partly because of the small heating surface and partly because of the very high temperatures required in the bottom vessel. The last effect, or upper vessel, was under atmospheric pressure.

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[3] McCulloh, Senate Document 209, 2nd Session, 29th Congress, p 82 (1847).
[4] Traite de Chaleur, 2, 300 (1828). Translated in McCulloh, *loc. cit.*
[5] "Fabrication du Sucre," 3d edition, Paris, 1911, p. 582.

About the same time, Degrande had the idea of multiple use of heat, and submitted his ideas to Derosne, of Maison Derosne & Cail (later Cail & Cie.). Degrande was retained by them, and his apparatus, often known as the Derosne apparatus, had some popularity. He connected a pan of the Howard or Roth type to a surface condenser consisting of a number of parallel horizontal tubes, arranged in series in a vertical stack.[6] Over this cascaded thin juice, and the resulting heating and atmospheric evaporation greatly increased the duty of the pan. There were two of these in operation in the United States in 1848.

Meantime Rillieux had returned to New Orleans, where in 1843 he built and patented (U. S. Pat. 3,237, Aug. 26, 1843) the first multiple effect evaporator operating under vacuum. In a letter dated 1844,[7] he states that it is fully successful, but that he does not wish his patent published till the apparatus is more fully worked out (a common procedure at that time). In 1847-8 [8] this system was fully described with drawings. Figure 35 is a copy of the original drawings of a 4-pan apparatus. Exhaust steam from the engines enters at I and rises through the right-hand column on body A. It passes into the tubes as shown. Vapors are drawn from the dome by pipe h; pass, by columns i and l, to the second effect, and so on. The boxes k, k[1] are condensate receivers. Body D is not a fourth effect, but receives steam through q and k, and therefore is in parallel with the second effect. It was used as a pan for final concentration. Columns p and r are connected by S to the vacuum pump.

Shortly after 1848 Brami Androae, an assistant of Rillieux's, sent drawings to Tischbein in Berlin, who sold them to Cail & Cie.; and patented them in Germany under his own name.[9] He misunderstood Rillieux's idea, and made a 3-pan apparatus, in which the center was the first effect, with a second effect on either side. He fed both second effects with liquid from the first. In Rillieux's 3-pan apparatus the boiling was only double-effect, but the third vessel was a finishing pan and was fed with thick juice. In Rillieux's 4-pan apparatus, the first three bodies operated as a triple effect, and the fourth was a finishing pan, boiling in double effect by using vapor from the first effect.

About this time Cail began to build vertical tube evaporators, under Tischbein's patents [10] and, according to Horsin-Deon, from Rillieux's ideas. The evaporator was still a double with three bodies, two second effects in parallel. Robert, director of the sugar factory at Seelowitz, Austria, is usually given credit for the vertical evaporator, but apparently the idea of upright tubes was Rillieux's. Robert, however, built

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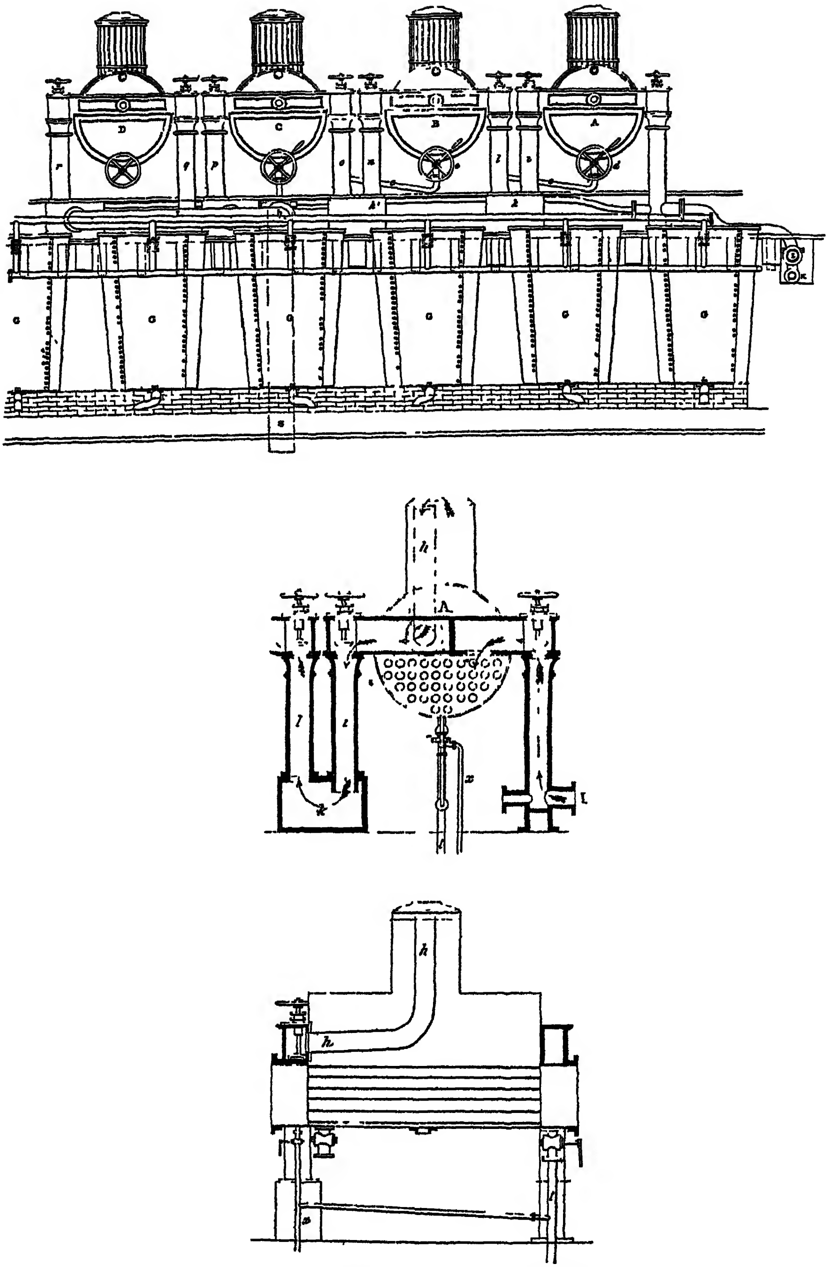
[6] U. S. Pat. Office Report, 1848, p. 322, McCulloh, *loc. cit.*, pp. 82-85.

[7] *Ibid.*, 1842-4, p. 295.

[8] *Ibid.*, 1848, pp. 328-32; McCulloh, *loc. cit.*, pp. 85-97.

[9] *Z. Ver. deut. Zuckerind.*, *1*, 139-47 (1851).

[10] *Ibid.*, *3*, 257-9 (1853).


D
C
B
A
r
q
p
o
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l
b
A'
G
S
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A
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k
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I
h
h

Fig. 35.—Rillieux's four-pan evaporator.

the first triple effect evaporator in Europe, and apparently the first straight triple ever built.[11]

Horizontal Tube Evaporators.

Under this heading will be discussed those evaporators having horizontal tubes with steam *inside* the tubes. Rillieux's original machine was of this type, with a body in the form of a horizontal cylinder. The first important development was in the '60's, when Brami Androae, Rillieux's assistant, developed the packing plate to permit easier removal of the tubes. Androae's first plate held six tubes in a circle around the stud, but otherwise was essentially the same as Figure 15.[12] During the period from 1850 to 1870 there was a drift toward vertical tube evaporators, and the next advance in the horizontal type came in 1879, when Hugo Jelinek and F. Wellner patented the modification still used and still known by their names.[13] In 1883, four years after the first machine was built, there were 237 bodies in operation with a total heating surface of 400,000 square feet.

Figure 36 is from the original patent drawings of the Wellner-Jelinek evaporator; and anyone familiar with present construction will see at once how little change has been made in 45 years. It differs from Rillieux in only three ways—the shape of the body, the baffles which lengthen the steam path, and the smaller diameter tubes. The body is made rectangular in plan with a dome-shaped top, to increase the boiling area, increase the cross-section of the vapor space, and give a greater height to avoid entrainment of material carried up by splashes. Liquor enters by g and leaves at i, and h is a wash-water valve. Steam enters by the valves l, and passes back and forth through the tubes as indicated, the cross-section of the later tube bundles being decreased to correspond to the smaller volume of steam entering them. Condensate is withdrawn at o, and the vents for non-condensed gases are not shown. The characteristic features of the Wellner-Jelinek type are the *trunk-shaped body and the multipass arrangement of the tubes.*

In the United States many firms have built evaporators of the

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[11] The history of evaporators is discussed in Koppeschaar, "Evaporation," pp. 18-30 (1913); Stohman-Schander, "Handbuch der Zukerfabrikation," 5th Ed., pp. 525-40 (1912); Greiner, "Verdampfen und Verkochen," pp. 2-18 (1912); but especially in Horsin-Deon, "Fabrikation du sucre de betterave," 3d ed. *I*, pp. 582-93 (1911); and Bolk, "Verdamping en Verdampinstallaties," *Arch. Suikerind., 17*, pp. 534-600, 609-707, 82 figs. (1909). The most valuable contemporary documents on the beginning of multiple effect development are McCulloh, "Report of Scientific Investigations Relative to the Chemical Nature of Saccharine Substances and the Art of Manufacturing Sugar," Doc. No. 209, Senate Documents, 2nd Session, 29th Congress, 1846-7; and Fleischmann, "Report on Sugar Cane, Its Culture, etc.," U. S. Pat. Office Report, 1848, pp. 274-338. It should be noted that McCulloh did not understand the internal construction of Rillieux's evaporator, and his drawings are incorrect.

[12] See also Witte, U. S. Pat. 1,373,041, 1921.

[13] *Z. Ver. deut. Zuckerind., 30,* 360 (1880); *32,* 302-5 (1882); German Pat. 6,958, 1879.

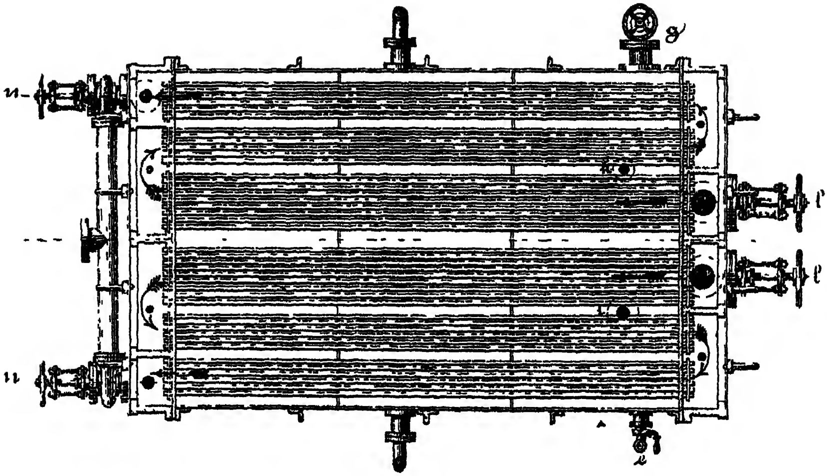


Fig 36A.—Wellner-Jelinek evaporator.

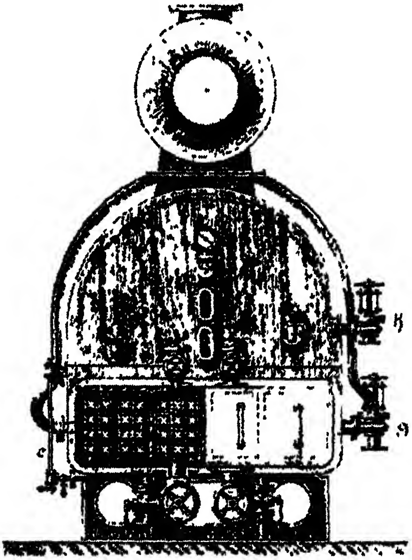


Fig. 36B.

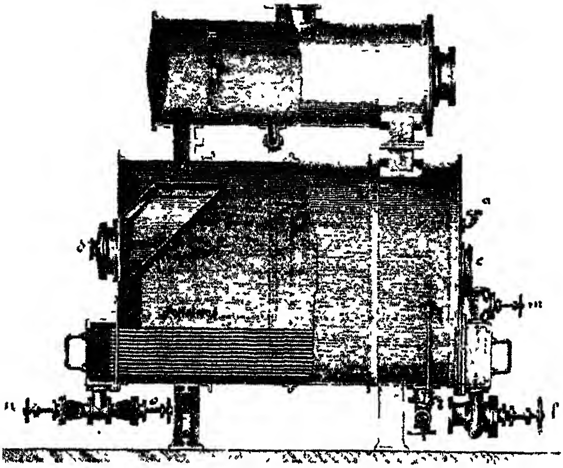
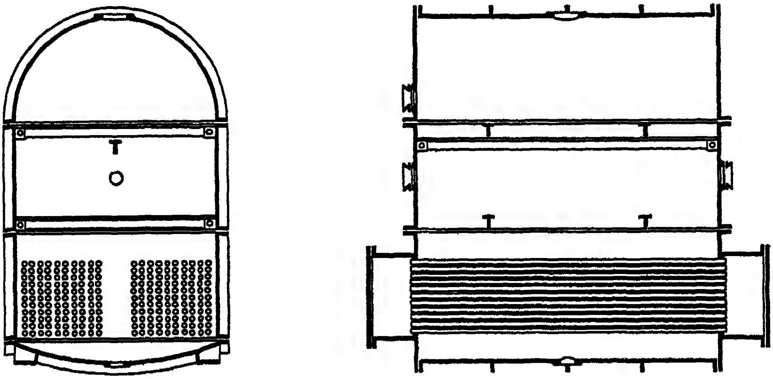


FIG. 36C.

Wellner-Jelinek type. Of the special forms which have been developed, we may mention the Swenson, the Zaremba, and the Buffalo types.

The Swenson Evaporator Company builds horizontal tube evaporators very like the old Wellner-Jelinek, except that their standard construction (see Figure 37) is of rectangular flat cast plates; and, in general (especially in the smaller machines), the sides are relatively higher, so that the body is more a vertical rectangular prism than the old Wellner-Jelinek. Practically all builders have abandoned the multipass steam travel of the original design. The Zaremba Company[14]



Courtesy Swenson Evaporator Co.

FIG. 37.—Standard Swenson horizontal tube evaporator.

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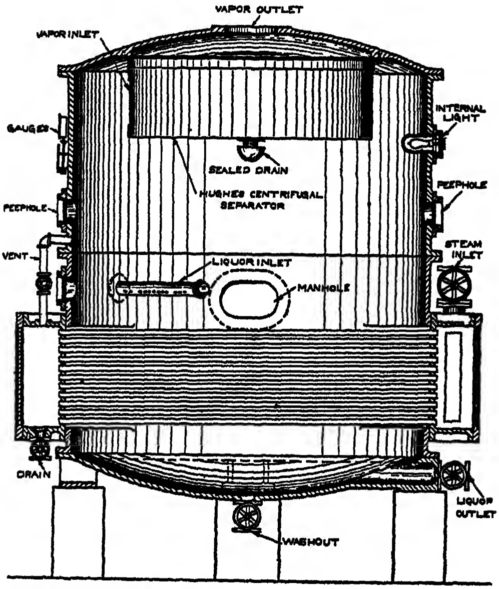
[14] U. S. Pat. 882,043, 1908.

builds horizontal tube evaporators with the body in the form of a vertical cylinder (Figure 38). This idea first appeared in an evaporator built by Simirensko in 1875.[15] The Buffalo Foundry & Machine Company builds horizontal tube evaporators with bodies in the form of a horizontal cylinder (Figure 39).

From a constructional point of view these forms are almost equivalent. The Swenson construction employs smaller castings than the others, and hence repairs are less expensive; and large machines can be built without the difficulty in shipments which may occur with cast rings in the Zaremba or Buffalo. On the other hand, the total length of joint and the number of corners where joints intersect is somewhat greater on the Swenson type than in the other two, and hence it may offer more opportunities for leaks In general, there have been few variations since Wellner and Jelinek in the horizontal tube evaporator; and such as there have been are rather unimportant.


VAPOR OUTLET
VAPOR INLET
GAUGES
INTERNAL LIGHT
SEALED DRAIN
HUGHES CENTRIFUGAL SEPARATOR
PEEPHOLE
PEEPHOLE
VENT
STEAM INLET
LIQUOR INLET
MANHOLE
DRAIN
LIQUOR OUTLET
WASHOUT

Courtesy The Zaremba Co.

Fig 38—Zaremba horizontal tube evaporator.

Vertical Tube Evaporators.

In this type the heating tubes are vertical, with liquor inside them and steam outside. Rillieux's and Robert's contributions have already been mentioned. The principal reason for the early shift to vertical tubes was the difficulty of cleaning the tubes of Rillieux's machine. The Robert type became so popular on the Continent that it was known as the "Standard" type, and this term is still used.

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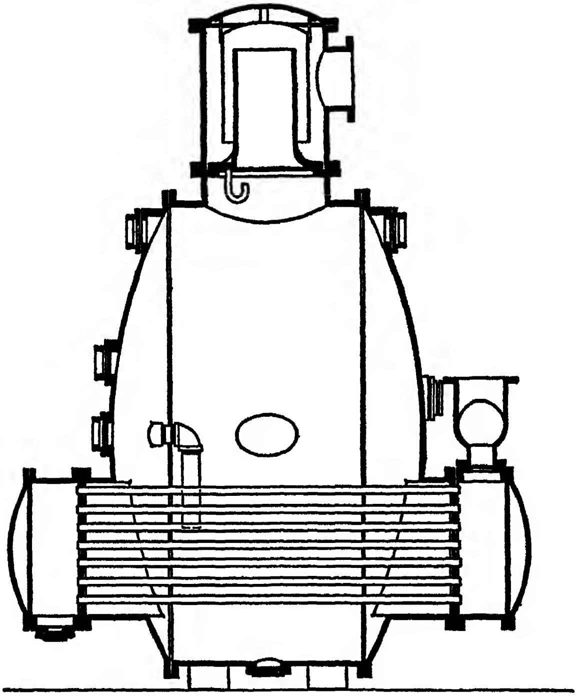
[15] Bolk, *loc. cit.*, p 572.

Figure 40 shows the original Robert vertical tube construction. Two horizontal tube sheets, A and B, divide the vertical cylindrical body into three sections. A number of tubes are expanded into the tube sheets, and steam is admitted at C. Connections may be made at any point in the lower tube sheet for draining condensate, and a connection is usually made in the top tube sheet for venting non-condensed gases. Vapor leaves the evaporator at D.



Courtesy Buffalo Foundry and Machine Co.

Fig. 39.—Buffalo horizontal tube evaporator.

The first fault to appear was poor circulation. The velocity up through the tubes was too high to permit any downward motion. Kasalovsky[16] was the first to introduce the central downtake (Figure 41). A wide tube E is placed in the center of the machine, and supplies a path for the return of liquid to the bottom of the tubes. In this form the vertical tube evaporator is still built, and thousands of such machines are in operation all over the world.

[16] Greiner, "Verdampfen und Verkochen," p. 17.

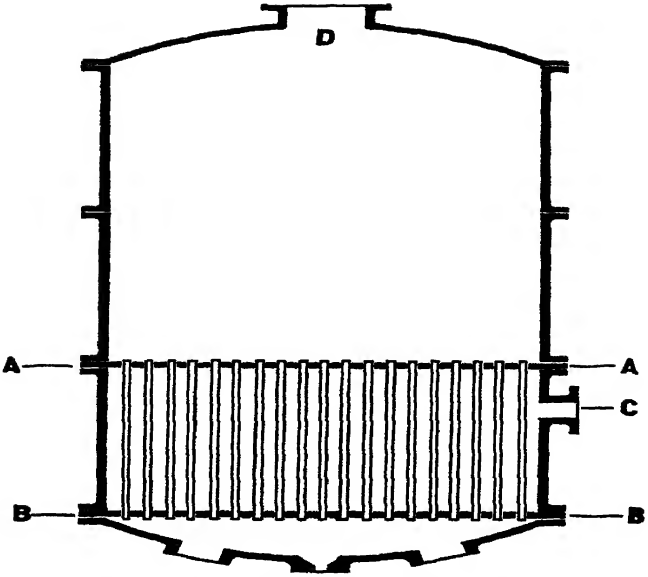


FIG. 40.—Robert evaporator.

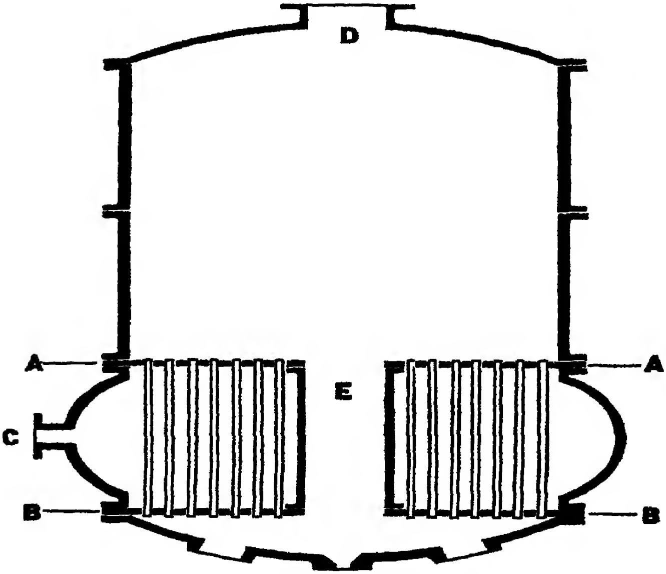


FIG 41.—Standard vertical tube evaporator.

In 1877 Riedel[17] built an evaporator (Figure 42) with an annular downtake and the heating surface in the form of a drum. This drum is usually carried on brackets; and steam may be supplied as shown or by a nipple connecting with the wall of the machine, or by a central pipe coming in from the top. This latter construction is easier to disconnect for repairs. These evaporators, usually known as the "basket type" are built by almost all makers in the United States. The principal advantage of this type is that the heating surface can be removed as a unit for repairs, or spare units kept on hand without undue expense. There has been considerable discussion as to whether or not the annular downtake or the central downtake is preferable from the standpoint of circulation, with the arguments and experience a little in favor of the annular downtake. No actual data under comparable conditions are available on this point.

In the early development of the vertical tube evaporator Walkhoff[18] hung a loose sheet steel cylinder inside the central downtake to separate, from the central core of descending liquid, the boiling layer on the walls of the downtake. This did not long survive and is practically never used. A similar outer cylinder has been used on the basket type.[19] The downtake has been placed at one side, instead of in the center of the body, and has been made of various shapes. One of the earliest ideas, before the idea of the central downtake, was to make some of the tubes larger than the rest, so that in these tubes there would be less pumping action and they could serve for the return circulation.[20] This survives in the Scott evaporator, which has a number of scattered downtakes instead of one central downtake. Other modifications will be discussed in Chapter 9.

Film Type Evaporators.

Three special designs will be mentioned here; not as the most important, but as representatives of widely different constructions not included in the above classes.

The *Kestner evaporator* (Figure 43) was first patented in 1899 by Paul Kestner of Lille, France.[21] It has been discussed in the technical literature more than any other form of evaporator.[22] It has vertical tubes with liquor inside and steam outside, but its distinguish-

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[17] Greiner, *loc. cit.*

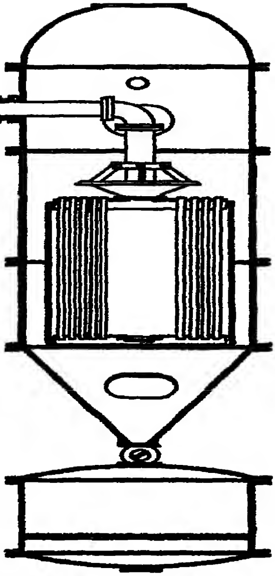
[18] Greiner, *loc. cit.*

[19] DeBeers, U. S. Pat. 1,006,363, 1911.

[20] Garrigues, U. S. Pat. 1,298,925, 1919.

[21] Kestner, U. S. Pat. 822,322, 1906 (reissued 13,117, 1910); 989,982, 1911; Parker, U. S. Pats. 940,473, 1909; 989,996, 1911; Kestner, Engl. Pat. 24,024, 1899.

[22] The more important references are: Reavell, *J. Soc. Chem. Ind., 37,* 172-87 (1918); Berner, *Z. Ver. deut. Zuckerind., 57,* 341-52 (1907); Klose, *ibid., 61,* 577-80, 832-40 (1911); Legier, *Int. Sug. J., 11,* 282-93 (1909); Van Troogen, *ibid., 11,* 578-82 (1909); Anon., *Engineering, 88,* 822-3 (1909); and a controversy in *Centr. Zuckerind., 18,* 1037, 1065, 1096 (1910).



Courtesy Swenson Evaporator Co.

Fig. 42.—Basket type vertical.

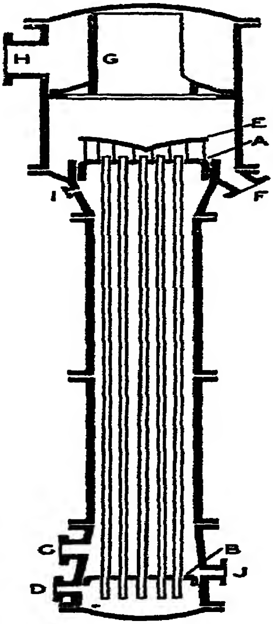

H
G
E
A
I
F
C
B
J
D

Fig. 43.—Kestner evaporator.

ing feature is the ratio of tube length to tube diameter—tubes are used up to 23 feet long and about 1¼ to 1½ inches in diameter. The static liquor level is carried only a short distance (3 to 5 feet) above the lower tube sheet.

Figure 43, A and B are the tube sheets, C is the steam inlet, and D the liquor inlet. As the liquor begins to boil, the steam carries it up into the tube and boiling becomes faster, the volume of steam increases and hence the velocity increases, until the liquid is actually carried the whole height of the tube by the steam velocity. The mixture of steam and spray issuing from the tubes strikes the baffle E, on the under side of which are spiral vanes. This gives the mass a whirling motion, which separates the liquid by centrifugal force, to be drawn off at F. The steam passes around further baffles G and leaves at H. Non-condensed gases are removed at I, and condensate at J. One feature to be noticed is that liquid being evaporated passes through any one body but once; therefore, feed must be accurately adjusted to correspond to the rate of evaporation if a uniformly dense discharge is to be maintained. A slight increase in steam pressure or slight increase in feed density might cause the tubes to go dry. Certain patents[23] provide for return of part of the discharge from the vapor head as recirculation, but this device is seldom used. When liquid

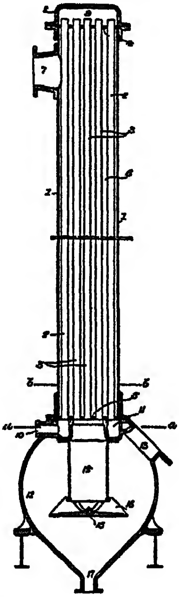


FIG. 44.—Kestner falling film evaporator.

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[23] Kestner, U. S. Pats. 1,028,737, 1912; 1,028,738, 1912.

comes into the feed chamber superheated (as frequently happens in multiple effect operation) the resulting flash makes difficulty in uniform distribution of liquid in the tubes, and a number of patents record devices to remedy this,[24] such as slots in the ends of the tubes where they project into the feed chamber, baffles in the feed chamber, etc.

The Kestner has also been developed into a falling film type.[25] Figure 44 shows the principle of the apparatus. Liquor is fed at 10, is distributed by the channel 11, rises through the outer tubes and descends as a falling film through the inner tubes. The steam inlet is shown at 7, the vapor[26] is taken off at 13, and liquor is discharged at 17.

A similar evaporator, with a separate pipe to return liquid from the head back to the feed box, and thus make control easier, has been patented by Webre[27] and has had considerable success.

The *Yaryan evaporator* (Figure 45) first patented by Homer T. Yaryan,[28] and later improved by C. Ordway,[29] is based on a principle similar to the Kestner but using horizontal tubes. The tube sheets, A and B, hold tubes of a convenient length. In the feed chamber C there may be partitions, or return bends, or some other device to connect three or five tubes into a continuous coil, with similar return bends located in the head at D. Limited amounts of liquid are fed into the first tube of each coil, and the process of carrying along a

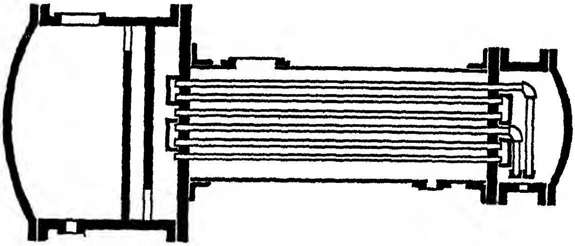

FIG 45—Yaryan evaporator (old style).

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[24] Kestner, U. S. Pats. 971,383, 1910; 1,005,553, 1911; 1,013,091, 1911, 1,022,321, 1912; Morris, U. S Pats. 971,394, 1910, 1,069,566, 1913; Mellor, U. S. Pat 991,342, 1911.

[25] Kestner, U. S. Pats. 965,822, 1910; 997,502, 1911, 997,503, 1911; 1,003,912, 1911; 1,005,553, 1911; 1,016,160, 1912; 1,060,607, 1913; 1,090,628, 1914; Parker, U. S Pat. 1,005,571, 1911.

[26] It is often convenient to distinguish between the steam used for heating a piece of apparatus, and the slightly lower-pressure steam generated by boiling liquid in the same apparatus, by calling the first "steam" and the second "vapor." This is not intended to imply any essential difference between the two. It is merely a convenient nomenclature.

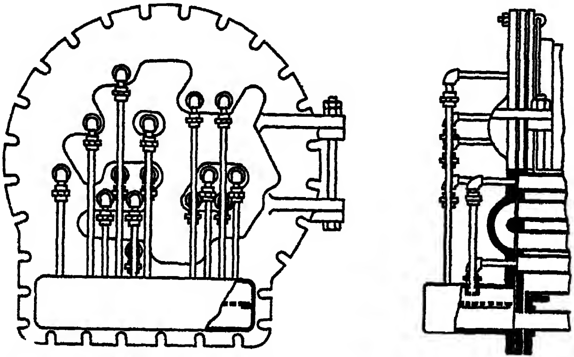
[27] U. S. Pat. 1,436,739, 1922.

[28] Anon, *Z. Ver. deut Zuckerind.*, *37*, 459-62 (1887); Springmuhl, *ibid*, *38*, 438-9 (1888); Anon., *ibid.*, *41*, 399-401 (1891); Yaryan, U. S. Pats 300,185, 1884; 355,289, 1886; 355,290, 1886; 383,384, 1888; 485,315, 1892.

[29] U. S. Pats. 714,513, 1902; 837,582, 1906; 942,407, 1909; 1,009,782, 1911.

film of liquid by the velocity of the steam is similar to that operating in the Kestner. The feed has been proportioned in various ways; the one shown in Figure 45 was generally used. Here all the feed tubes extended down to a common level and were closed by a cap or plug carrying a drilled orifice. For small machines, the difference in level between these orifices and the inlets of the various coils was not large compared to the total pressure drop through the apparatus. Hence the variations in head on the feed orifices to coils in different positions were not serious, and feed was fairly uniformly distributed to the coils. For large evaporators this is not true, and sometimes the feed chamber had to be divided into two or more parts to give satisfactory feed in the top tubes. In the head are baffles E to separate steam from spray. Liquor is drawn off at F and the vapors escape at G.

A more recent development is shown in Figure 46, in which the tube sheet is prolonged at the lower end and the feed pipe connected to it. A casting is hinged to the frame of the machine and carries a feed distributing box and feed pipes. The return bends are cast as an integral part of this plate. The construction makes the cleaning of the tubes simpler, and allows control of the feed from outside.



Courtesy Swenson Evaporator Co.

FIG. 46.—Yaryan feed end, new style.

The Yaryan is especially successful with foaming liquors, such as soda pulp liquors, glue, gelatin, etc. Another advantage of the Yaryan is that the use of return bends permits long liquor paths without using tubes of unusual length. The commonest form of Yaryan uses three passes to the coil, and not over 20 foot tubes to the pass. The tubes are often shorter than 20 feet. This, together with its horizontal arrangement, makes tube replacement simple, yet gives a long liquor path and high velocities.

The Mirrlees-Watson Company, of Glasgow, build the Yaryan evaporator in a slightly different form.[30] The heads of the several bodies are united into a column which is a unit structurally, and contains the partitions necessary to make the apparatus function as a multiple effect evaporator.

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[30] Engl. Pats. 11,485, 1888; Watson and Robinson, 17,809, 1888; 8,790, 1890; Robinson and Ballingall, 15,698, 1890; Mirrlees, 8,809, 1891.

The *Lillie evaporator*[31] was invented by S. M. Lillie of Philadelphia in 1890 (Figure 47). There is but one tube sheet, A, thickened in its central portion, into which tubes are expanded. The tubes are short enough and their bearing in the tube sheet is long enough so that they are self-supporting. Steam enters at B, and fills the tubes. A disc is rolled into the right-hand end of each tube, and in it a hole is drilled to act as a vent for non-condensed gases. At the bottom of the body is a centrifugal pump E, discharging by pipe F to the next effect or to a receiver for thick liquor, and by pipe G, to distributing tubes. The liquor cascades over the heating tubes and is evaporated in thin films. The vapors leave at H. A float-valve I controls the feed inlet to the body, and therefore the proportion of the pump discharge that is fed from effect to effect in a multiple effect apparatus is automatically controlled by the draw-off from the last one. Thus the Lillie is free from the difficulties of control such as the Kestner or Yaryan experience, where a given amount of liquid passes through the body only once.[32] A similar evaporator has recently been patented by Hughes.[33]

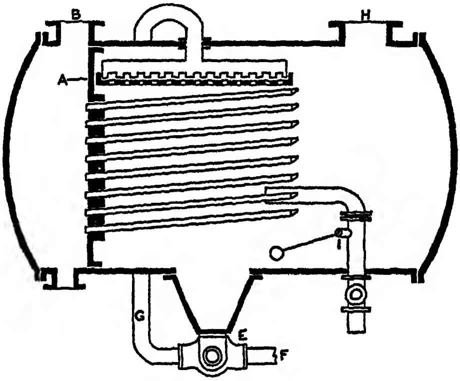

Fig. 47.—Lillie evaporator.

There are other commercial types of evaporator bodies than described above and many minor modifications of those mentioned. These either developed from a consideration of some factor affecting heat transfer, and hence will be discussed in Chapter 9; or they were developed for some special use or special industry, and will be described in Chapter 13.

[31] U. S. Pats 422,235, 1890; 440,231, 1890; 498,938, 1893; 521,215, 1894; 740,449, 1903; 789,159, 1905; 939,143, 1909; 988,477, 1911.

[32] Lillie, U. S. Pat. 491,659, 1893.

[33] U. S. Pat. 1,506,001, 1924.

Chapter 9.

Heat Transfer in Evaporators.

From the considerations of Chapters 3 to 6, we see that in the ordinary case of heat transfer in evaporators there are three resistances: a film resistance on the steam side, a film resistance on the water side, and a resistance due to the metal of the tube. If scale is present, it forms a fourth resistance. The overall rate of heat transfer is, then,

$$U = \frac{1}{\frac{1}{K_1} + \frac{d_1}{\lambda_1} + \frac{d_2}{\lambda_2} + \frac{1}{K_2}} \qquad (27)$$

where K_1 is the steam film coefficient, K_2 the water film coefficient, d_1 and d_2 the thicknesses of tubes and scale, and λ_1 and λ_2 the thermal conductivities of metal and scale, respectively. The data of Chapter 6 show that in no case can we calculate U, as data are almost entirely lacking for K_1 and are totally lacking for K_2. We may, however, discuss qualitatively the factors that affect each of these, and the manner in which such effects are interpreted in actual design.

Hydrostatic Head.

Before we discuss the factors affecting coefficients, mention should be made of two general factors affecting capacity through their effect on temperature drop. These are hydrostatic head and boiling point elevation.

The capacity of any evaporating apparatus is obviously measured by the total heat transferred according to the equation

$$Q = UH\Theta_m \qquad (51)$$

where Q is the total heat transmitted per hour, U is defined by equations (26) and (27), H is the heating surface, and Θ_m the mean temperature difference between steam and liquid.

If a vessel contains a boiling liquid, and the space over the vessel has a pressure p, the liquid *at the surface* will boil at a temperature t corresponding to this pressure. But a particle at a distance h below the surface of the liquid is under a pressure $p_1 + h$ (if p_1 and h are both measured in terms of head of the liquid being boiled) and consequently cannot boil till it reaches a temperature t_2 corresponding to the increased pressure. In an evaporator of commercial size, boiling vigorously, a thermometer inserted in the lower part of the liquor space may show readings the same as, or only a little higher than, the liquid at the surface. The layers of liquid immediately next the heating surface must come to this higher temperature t_2 before they can boil, though as soon as vapor forms the superheated liquid is probably stirred into the main stream, to give off its superheat by mixing with colder liquid or by flashing as it comes nearer the surface.

The effect is not serious at atmospheric or higher pressures, but it becomes very serious at low pressures. Consider a standard vertical tube evaporator, boiling a liquid of specific gravity 1.0, with tubes 6 feet long. A particle at the bottom of the tubes is under a hydrostatic head of 6 feet of liquid, or $\frac{6 \times 12 \times 25.4}{13.6} = 135$ mm. of mercury. If the liquid is boiling at 212° at the surface, ($p_1 = 760$ mm.) $p_1 + h = 895$ mm. which corresponds to a boiling point of 220.3° F., an elevation of 8.3°. If the liquid at the surface is under a vacuum of 26 inches (absolute pressure 100 mm. Hg) the boiling point at the surface is 124.9° F., the pressure $(p_1 + h)$ is $100 + 135 = 235$ mm., and the boiling point t_2 is 158.2°, an elevation of 33.3° F. If the steam in the first case were 10 pounds gage (239° F.), the mean temperature drop would not be $(239 - 212)$ or 27° F., but $239 - \frac{(212 + 220.3)}{2}$ or about 23° F., a loss of 15 per cent. If the steam in the second case were also at 10 pounds gage (single effect operation), the mean temperature drop would not be $(230 - 125) = 114°$ F., but $239 - \frac{(125 + 158)}{2}$[1] or 98° F., again a loss of 15 per cent. If, however, the vessel boiling under a 26-inch vacuum were the last effect of a multiple effect evaporator, receiving steam from the preceding effect at 170° F., the temperature drop would be, not $(170 - 125)$ or 45°, but $170 - \frac{(125 + 158)}{2} = 29°$, a loss of 35 per cent. Hence, the smaller the working temperature drop and the higher the vacuum, the more important hydrostatic head becomes. If the liquid has a specific gravity greater than 1.0, the effect is proportionately increased.

Such calculations as the above are uncertain for two reasons. First, when the liquid is boiling, the tube is filled with a mixture of steam and liquid, not a solid column of liquid. This is particularly true if the static level is below the tops of the tubes. In this latter case, if there is an unbroken film of liquid to the top of the tube, the full hydrostatic head will be exerted on the lower layers irrespective of the apparent static level. On the other hand, if the film is completely broken by bubbles at any point, or if it is so thin that it is held in place by capillary or interfacial forces and does not flow under the influence of gravity, then the upper parts of the film do not exert any hydrostatic head, and the elevation of boiling point at the bottom of the tube is due to static head only. It is often argued that if the tube be filled with a column of liquid containing steam bubbles, the hydrostatic head will be decreased because of the decreased mean weight of the liquid. Claassen himself fell into this error.[2] That it is an error is easily seen by considering that the cross-section of a vertical column of liquid has no bearing on the hydrostatic head at the bottom; a constriction does not decrease the pressure. Hence, the parts of the column of liquid around the bubbles transmit the head undiminished.

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[1] The arithmetic mean is not strictly correct here, as temperature is not directly proportional to pressure. It is amply accurate for the present purpose.

The second uncertainty is due to the question of whether or not any of the liquid is ever heated to this higher temperature corresponding to the increased head at the bottom of the tube. Except for the most viscous and sluggish materials, it is certain that the mass of the liquid is little, if any, hotter at the bottom of the tube than at the top. But we must consider that there is a stagnant liquid film, even with boiling liquids, and this film must attain the calculated temperature, except with the most vigorous artificial circulation, and hence its temperature determines the *working* temperature drop at this point.

That the two arguments advanced in the above paragraphs are not substantiated in practice is indicated by the harmonious results obtained, in the following sections, by recalculating experimental work on the assumption that the high temperatures called for by hydrostatic head really exist in the films.

Experimental Data on Hydrostatic Head.

Experimental work to cover this point is not entirely conclusive, as changes in level always bring corresponding changes in velocity of circulation, and the results of experiments therefore show the combined effect of these two factors. Wellner and Jelinek appreciated the effect of hydrostatic head and used it as an argument for their type. Claassen[3] pointed out that the apparent rate of heat transfer was higher when the liquor level was low. Kerr,[4] and Badger and Shepard[5] have published experimental data on vertical tube evaporators, and Badger[6] has published data on a horizontal tube evaporator. The results of these last two sets of experiments are shown in Figures 48 and 49. Figure 50 contains some unpublished data from the horizontal tube evaporator, expressed in a different form. The data of Figure 48 are from a vertical evaporator of the basket type, 30 inches inside diameter, with an 18-inch diameter steam basket containing

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[2] *Z. Ver. deut. Zuckerind., 42,* 550-60 (1892).

[3] *Z. Ver. deut. Zuckerind., 43,* 236-69 (1893).

[4] Bull. 149, La. Agr. Exp. Sta., pp. 27-31; *Met. Chem. Eng., 11,* 333-5 (1913).

[5] *Trans. Am. Inst. Chem. Eng., 13* (I), 139-49 (1920); *Chem. Met. Eng., 23,* 390-3 (1920).

[6] *Trans. Am. Inst. Chem. Eng., 13* (II), 139-53 (1921); *Chem. Met. Eng., 25,* 459-62 (1921).

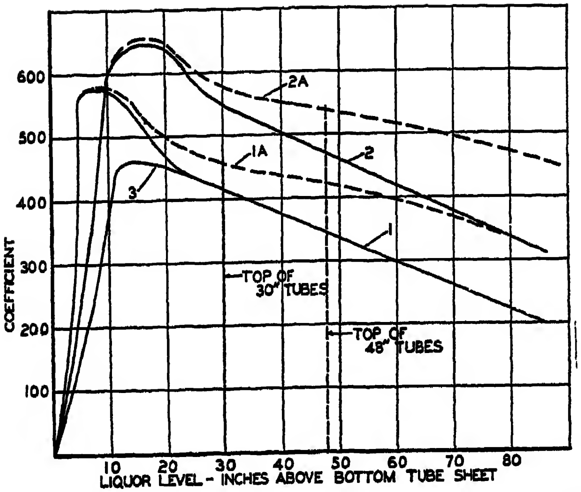


FIG 48—Effect of hydrostatic head on vertical tube evaporator. (See Fig. 57)

Curve 1—Tubes 30" long, flat bottom, apparent coefficients.
Curve 1A.—Tubes 30" long, flat bottom, corrected coefficients.
Curve 2—Tubes 48" long, flat bottom, apparent coefficients.
Curve 2A—Tubes 48" long, flat bottom, corrected coefficients
Curve 3—Tubes 30" long, cone bottom, apparent coefficients.

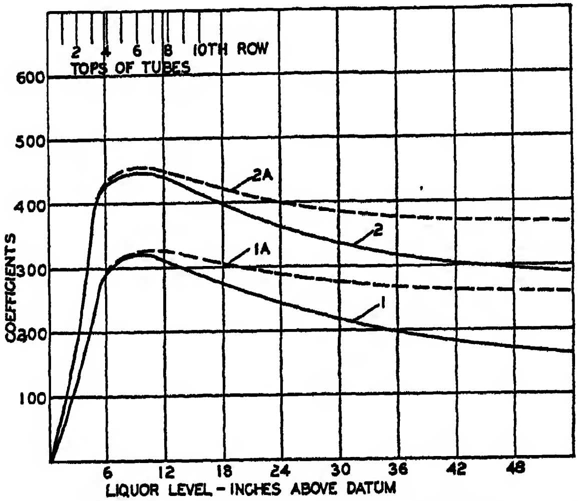


FIG. 49—Effect of hydrostatic head on horizontal tube evaporator.

Curve 1—Temperature drop 15° C, apparent coefficients
Curve 1A.—Temperature drop 15° C, corrected coefficients.
Curve 2—Temperature drop 25° C., apparent coefficients.
Curve 2A—Temperature drop 25° C., corrected coefficients.

twenty-four 2-inch iron tubes. For Curves 1 and 1A, they were 30 inches long; for Curves 2 and 2A, they were 48 inches long. Figures 49 and 50 are data from a horizontal tube evaporator 26 inches wide by 46 inches long inside, fitted with ⅞-inch steel tubes. In Figure 49 there were 156 tubes in 10 horizontal rows; in Figure 50 there were 124 tubes in 12 horizontal rows. All the work in Figures 48 and 49 was done at a boiling point of 167° F. (75° C.); the work in Figure 50 was at 140° F. (60° C.). Figure 49 contains two curves, each having a constant temperature drop, but a variable level. Curves 1 and 1A were run with a 27° F. (15° C.) temperature drop; Curves 2 and 2A, 45° F. (25° C.). In Figure 50, each curve had a constant level but a varying temperature drop. Curves 1 and 1A, Figure 50, were run with the level 0.5 inch over the top tubes, Curves 2 and 2A, 12 inches over the tubes, and Curves 3 and 3A, 24 inches.

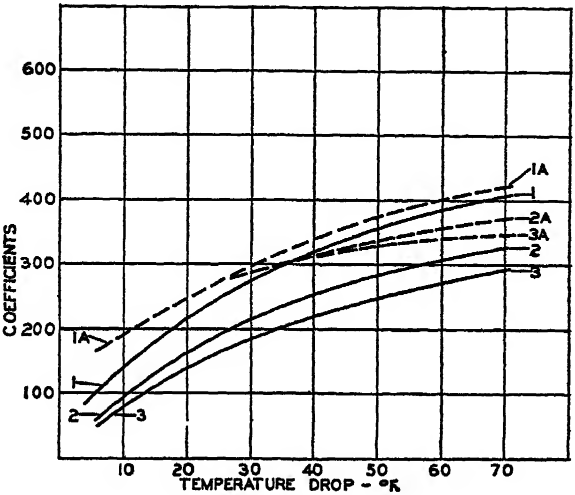


Fig. 50.—Effect of hydrostatic head and temperature drop on horizontal tube evaporator.

Curve 1.—Level 0.5" above tubes—apparent coefficients.
Curve 1A.—Level 0.5" above tubes—corrected coefficients.
Curve 2.—Level 12" above tubes—apparent coefficients.
Curve 2A.—Level 12" above tubes—corrected coefficients.
Curve 3—Level 24" above tubes—apparent coefficients.
Curve 3A.—Level 24" above tubes—corrected coefficients.

In these figures the solid lines show the rate of heat transfer calculated by using the difference between steam temperature and vapor temperature, both calculated from pressures. This difference is called the *apparent temperature drop.* As discussed in the preceding section, this does not give the true mean temperature drop; but the coefficient so calculated has a value in commercial design, so it is often used and called the *apparent heat transfer coefficient.* The dotted lines show the coefficients calculated on the true mean temperature drop as discussed in a preceding paragraph; and should therefore be independent of hydrostatic head The fact that they do change with change in level shows that changes in level affect other factors than temperature drop (principally velocity of circulation). The effect of hydrostatic head is essentially the vertical distance between the solid and the dotted curve in each case The experiments of Figures 48 and 49 were run with medium boiling points (167° F.); data taken at lower boiling points would show a much greater effect. These data also show the relatively small effect of hydrostatic head in the horizontal evaporator, except at excessively high levels.

Effect of Hydrostatic Head on Design.

The Robert or vertical tube evaporator early became the standard type in Europe, but it was well recognized that it would have a higher capacity if, instead of being full of liquid, only a thin film on the heating surface was permitted. It was not clear at this time whether or not the effect desired was due to elimination of hydrostatic head, increase in liquor velocity, or some other factor. This led to the development of *"riesel"* evaporators,[7] which may be translated as "trickling" evaporators. In this system, devices were provided at the upper ends of the tubes whereby only a film of liquid flowed down inside the tube. There was no circulation in the ordinary sense; all the evaporation in any one body was accomplished by the liquor passing down the tubes once.

The literature from 1889 to 1896 is full of descriptions of such devices and discussions of their advantages and disadvantages[8] Many devices were used, usually some sort of thimble or cap on the upper end of the tube,[9] or rods carrying shelf-like projections[10] or the liquid to be evaporated was injected tangentially under pressure.[11] Later Claassen suggested "trickling from below" (*rieselung von unten*) by carrying a low liquor level, and also hanging wood rods in the tubes to decrease the free cross-section and increase the velocity.[12] The same idea was applied to horizontal evaporators by showering the liquid over the tubes by a perforated plate.[13] These various devices were rather widely used, and increased capacities of 40 to 100 per cent were

[7] This system is called *"russeilment"* in France.

[8] For instance, Scheller (and discussion), *Z. Ver. deut. Zuckerind*, *42*, 550-60 (1892); Ehrhardt, *ibid*, *43*, 59 (1893); Wagner, *ibid.*, *43*, 640 (1893).

[9] Greiner, Germ. Pat. 58,037, 1890, *Z. Ver. deut Zuckerind*, *42*, 131-2 (1892); Schwager, 74,877, 1892; *ibid.*, *44*, 511-2 (1894); Althoff, 129,038, 1901; *ibid.*, *52*, 268 (1902); Bouvier, *ibid*, *42*, 871-3 (1892).

[10] Schroeder, Germ. Pat. 61,275, 1891; *ibid*, *42*, 343-4 (1892).

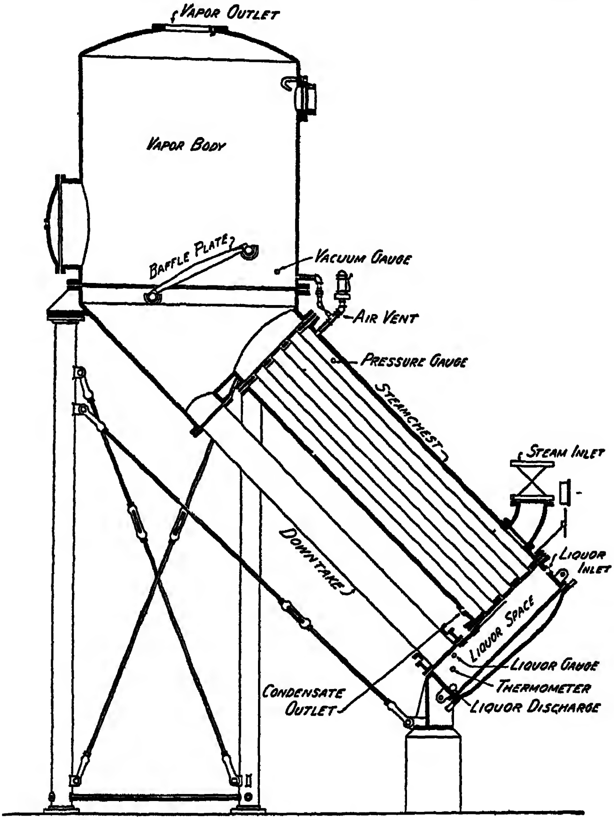
[11] Einsporn, Germ. Pat. 75,384, 1893; *ibid.*, *44*, 814 (1894).

[12] Claassen, *Z. Ver. deut. Zuckerind.*, *45*, 1-3, Battut, *ibid.*, *45*, 513-7; Claassen, *ibid*, *45*, 517-21 (1895); Mueller, *ibid.*, *49*, 512-3 (1899). See also p. 137.

[13] Meyer, U. S Pat. 780,612, 1907; Haacke and Schallehn, Germ. Pat. 65,287, 1892; *Z. Ver. deut. Zuckerind.*, *43*, 528 (1893).

reported as a result. Needless to say, all these devices have by now totally disappeared. It should be noted that they were used at a time when vertical evaporators were usually made without downtakes. Modern constructions give vigorous enough circulation without the use of such very troublesome devices.

It is interesting to note that both the Lillie and the Yaryan evaporators appeared during this period, and are frequently mentioned in discussions of trickling or film evaporators. The Lillie is solely a device for eliminating hydrostatic head. The Yaryan does eliminate it, though it is rather a device for increasing liquor velocity.


Vapor Outlet
Vapor Body
Baffle Plate
Vacuum Gauge
Air Vent
Pressure Gauge
Steamchest
Steam Inlet
Liquor Inlet
Downtake
Liquor Space
Liquor Gauge
Thermometer
Liquor Discharge
Condensate Outlet

Courtesy Buffalo Foundry and Machine Co.

Fig. 51.—Buflovac inclined evaporator.

In many ways it is desirable to have liquid inside the tubes instead of outside, primarily for ease in cleaning. A strictly horizontal machine of this type would be the Yaryan. Many other designs have been proposed with the tubes somewhat inclined. This permits circulation within the effect. One of the first was the Brandt evaporator,[14] which is much like a Babcock-Willcox water-tube boiler. It has an inclined drum containing tubes, heated by steam, with liquor boxes at either end. Above this is a horizontal drum receiving liquor from the upper end of the tubes, and feeding it back to the lower end. Another design is that of Prache and Bouillon.[15] A similar type is built in this country by the Buffalo Foundry and Machine Company (Figure 51). The method of operation is obvious, the larger lower tube serving as a return for the circulation.

All the devices, except the standard horizontal, that operate with diminished hydrostatic head, derive their major advantages from increased circulation, and will be further discussed under that head.

Boiling Point.

It should not be necessary to discuss such an elementary consideration as the effect of vacuum, but there is a surprisingly widespread opinion that a vacuum evaporator is mysteriously and magically different from an apparatus working under pressure. There is only one reason for employing a vacuum—to lower the boiling point. And this in turn is done for one of two reasons: to increase the available temperature drop, or to protect a substance sensitive to elevated temperatures (such as glue, fruit juices, etc.).

Five-pound exhaust steam has a temperature of about 225° F. and a total heat content of 1155 B.t.u. If this is used to boil water at atmospheric pressure, and if the condensate may be cooled to the temperature of the boiling liquid, we will have an available temperature drop, between steam and water, of 13° F., and we lose 180 B.t.u. per pound in the hot condensate, leaving 975 B.t.u. available per pound of steam. If we boil under a vacuum of 26 inches, the temperature of the water is about 125° F., and the heat lost in the condensate is 93 B.t.u. Hence, under vacuum there is an available temperature drop of 100° instead of 13°, and the total available heat per pound of steam is 1062 B.t.u. The capacity of the apparatus under vacuum, assuming equal heat transfer coefficients, is $\frac{100}{13}$ or 7.7 times as great as when working at atmospheric pressure; and the amount of steam used is $\frac{975}{1062}$ or 0.92 as much. The saving in steam is not significant, the increase in capacity is. Due to lowering of coefficients at low boiling points, the actual increase in capacity is less than the theoretical.

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[14] Kuhl, *Sucr. Belge*, *42*, 297-300 (1914); Molenda, *Z. Zuckerind. Böhm*, *43*, 304-21 (1919); Hallstrom and Brandt, Germ. Pats. 265,675, 1911; 265,677, 1911; 281,720, 1913; Brandt, Fr. Pat. 437,997, 1911.

[15] *J. Soc. Chem. Ind*, *36*, 72 (1917); U. S. Pats. 972,572, 1910; 1,071,341, 1913; Engl. Pats. 25,458, 1912; 9,276, 1908, Germ. Pat. 267,270, 1911.

Rise in Boiling Point of Solutions.

A factor that is frequently neglected is the difference between the boiling point of a solution and the boiling point of water at the same pressure. This elevation is often discussed in works on physical chemistry, and for *dilute, ideal* solutions it is usually considered that the rise in boiling point is proportional to the molecular concentration and independent of the substance dissolved. This may be expressed as

$$\Delta T = \frac{m}{M} K_b$$

where ΔT is the rise in boiling point caused by dissolving m grams of a substance of molecular weight M in 1000 grams of the solvent. Where the solvent is water, K_b has the value 0.935° F. For instance, a saturated sodium chloride solution, boiling at atmospheric pressure, contains 42.0 parts of salt per 100 parts of water. Therefore, $m = 420$, $M = 58.46$, and ΔT should be 6.7° F. Washburn[16] gives a formula based on thermodynamic reasoning:

$$\log_{10} x = -\frac{0.4343 L_v \Delta T_B}{R T^2_B}$$

where x is the concentration expressed as gram-molecules solvent per gram-molecule of (salt + solvent), L_v is the latent heat of evaporation in calories per gram-molecule, R is the gas constant (1.98), T_B is the boiling point of the solvent in degrees absolute (° C. + 273.1), and ΔT_B is the elevation in boiling point of the solution, in degrees C. If this formula be applied to the saturated sodium chloride solution mentioned above, it gives 3.5° C., or 6.2° F. The actual elevation of boiling point of this solution is 16.1° F.

The above discussion must not be interpreted as a criticism of the value of work done by physical chemists in this field. To derive any theoretical results at all, it is necessary to make certain thermodynamic assumptions that do not hold for strong solutions of inorganic salts in water. The theory of strong solutions is such a difficult field that practically no progress has yet been made in it. For the present we must rely on empirical data; which are, however, almost entirely lacking.

The most useful empirical generalization yet made is that of Baker and Waite,[17] who showed that Dühring's rule is quite rigidly applicable to solutions. This idea has been further developed by Leslie and Carr.[18] Dühring's rule as applied to solutions may be expressed as

$$\frac{t_1 - t_2}{\Theta_1 - \Theta_2} = K \tag{52}$$

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[16] "Principles of Physical Chemistry," p. 200 (1921).

[17] *Trans. Am. Inst. Chem. Eng.*, *13* (II), 223-32 (1921); *Chem. Met. Eng.*, *25*, 1137-40 (1921).

[18] *Ind. Eng. Chem.*, *17*, 810-17 (1925).

where t_1 and t_2 are the boiling points of a solution at two different pressures, and Θ_1 and Θ_2 are the boiling points of the solvent at the same two pressures In other words, if along one coordinate are plotted the boiling points of the solvent, and along the other the boiling points of the solution at the same pressures, the points for any one solution fall on a straight line. Figure 52 shows the Duhring lines for sodium chloride.[19] Up to the present, the only full data so reported are sodium chloride, calcium chloride,[20] and glycerine with and without

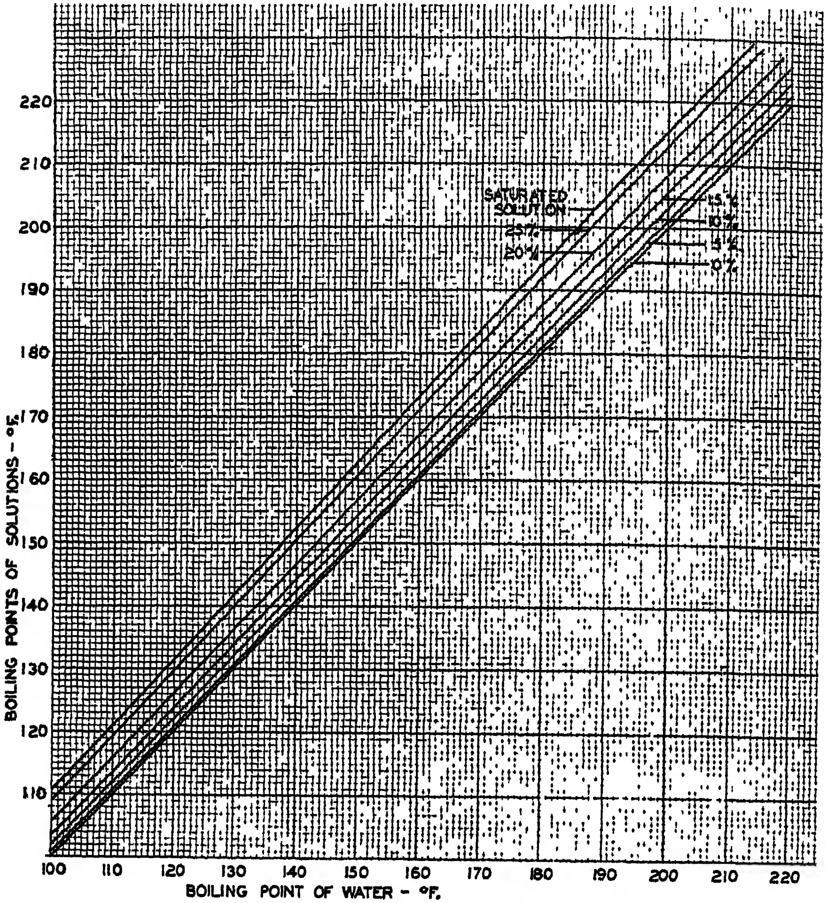

FIG. 52.—Boiling points of sodium chloride solutions (Duhring lines).

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[19] From data given by Badger and Baker, *Trans. Am. Inst. Chem. Eng.*, *13* (I), 151-68 (1920); *Chem. Met. Eng*, *23*, 569-74 (1920).

[20] Baker and Waite, *Trans Am Inst. Chem. Eng.*, *13* (II), 233-42 (1921); *Chem. Met. Eng.*, *25*, 1174-8 (1921).

sodium chloride.[21] This relationship does not enable us to predict the boiling point of a solution, but it greatly simplifies both the work of determining the boiling points of solutions over a range of pressures, and the representation of the final result.

The application of this to evaporator design is obvious at once. An evaporator boiling saturated sodium chloride solution at atmospheric pressure with steam at 240° F. will have, not (240 — 212) or 28° temperature drop, but (240 — 228) or 12° temperature drop. In the case of many substances which are more soluble than common salt, such as caustic soda, calcium chloride, etc., the losses may be many times as great. In the case of organic materials of high molecular weight, or colloidal solutions (paper pulp liquors, packing house tankage, glue, etc.) the elevation in boiling point may be so small as to be of no practical significance.

Since Faraday's time there has been a discussion as to whether or not the steam arising from a solution was superheated to the temperature of the solution, or saturated at the pressure in question and hence colder than the solution. This question has recently come to a head in a number of papers [22] and it is now quite certain that the vapor is superheated. In multiple effect operation this results in a serious loss. It will be shown later that the temperature of such superheated steam is of no significance in evaporator operation, and that the temperature of saturation determines the working temperature drop. Thus, in the evaporator above, we lose 16° of available temperature drop when boiling at atmospheric pressure. If the steam evolved by the solution is to be used to heat a later effect, the fact that it is superheated to 228° is of no significance. The significant fact is that this vapor is at atmospheric pressure, and therefore it gives up the major part of its heat only after it has cooled to 212°, and we therefore lose this elevation in boiling point for *every* effect. This seriously limits the number of effects that may be used for concentrating solutions with a large elevation of boiling point.

If a solution is being continuously concentrated in a vessel, the whole vessel must be filled with material at the final density, and hence of the *final* boiling point. Attempts have been made to design evaporators to minimize this effect. Prache and Bouillon [23] suggest the construction shown in Figure 53. The liquor space below the lower tube sheet is divided into several compartments, r, and corresponding partitions, p, extend some distance above the upper tube sheet. Each compartment has its own downtake tube n. Near the bottom of each downtake is a smaller pipe o, opening upwards into the downtake and connecting with the next lower compartment. Liquor is fed into the first bottom compartment at v, and circulates up through the tubes and back by the downtake in the usual way. A part is diverted by pipe o to compartment 2, and so on. Since the flow from compartment to compartment depends only on a difference in head between compartments, the amount of thick liquor drawn off from the last compartment at S regulates the amount fed through all the others.

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[21] Carr, Townsend, and Badger, *Ind. Eng. Chem.*, *17*, 643-6 (1925).

[22] Schreber, *Z. Tech. Phys.*, *4*, 19-27 (1923); *Chem. App.*, *10*, 169-70, 177-8, 158-9, 193-5 (1923); *Chem. Met. Eng.*, *31*, 297-300 (1924); Claassen, *Centr. Zuckerind.*, *32*, 390 (1924); Bahlke and Wilson, *Chem. Met. Eng.*, *32*, 327-9 (1925); Discussion, *Chem. Met. Eng.*, *32*, 387-8 (1925).

[23] Schroeder, *Chem. App.*, *2*, 187-8 (1915); U. S. Pat. 896,460, 1908; Engl. Pat. 26,065, 1905; Germ. Pat. 180,115, 1905; Fr. Pat. 364,408, 1906.

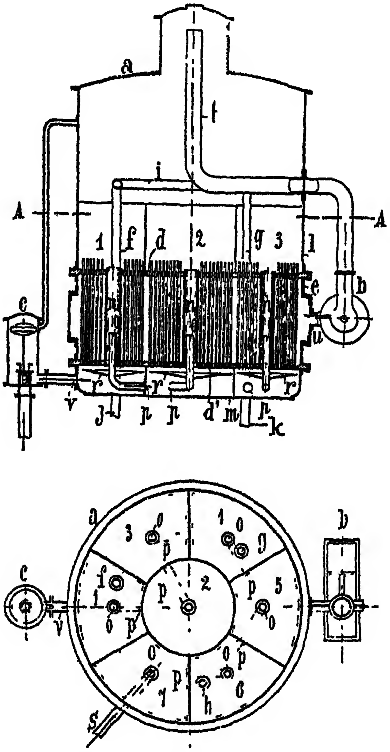

FIG. 53—Prache and Bouillon vertical evaporator with compartments.

The interior of the steam space is not divided. Thus, instead of the whole heating surface being under the handicap of the elevation in boiling point of the final solution, only a part of it operates with this small final temperature drop. The other compartments have progressively higher drops back to the first, which has the largest. A similar idea is incorporated in the Vincik-Turek evaporator[24] which has recently received much publicity in Czecho-slovakia. This type of construction is only necessary where the evaporator must work under very small temperature drops, as in thermocompression, or pressure evaporators for sugar solutions. Later reports[25] indicate that it is not necessary for the second purpose.

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[24] Koppl, *Z. Zuckerind Cechoslov. Rep.*, *47*, 395 (1923); Linsbauer, *ibid.*, *47*, 574 (1923).

STEAM FILM COEFFICIENT.

The factors which may affect the steam film coefficient have been discussed in general in Chapter 6. Here, we will attempt to apply them more specifically to evaporators, though there is practically no quantitative data available. We should consider

Steam density.
Steam velocity.
Steam richness.
Superheat in the steam.
Condensate temperature.
Condensate drainage.
Deposits on the steam side of the tubes.

There is no evidence bearing directly on the effect of *steam density.* The only experimental data available are *overall* coefficients, which include effects on the liquor side. If the boiling point be held constant and the steam pressure increased, the resulting increased temperature drop causes more vigorous boiling and better circulation on the liquor side, and hence results are not comparable. If the temperature drop be held constant, while the boiling point and the steam pressure are both increased, the decreased viscosity of the liquid at higher temperatures again causes an increase in the liquid film coefficient and clouds the issue. Until *steam film coefficients* are available we cannot tell whether or not an increase in steam density increases the steam film coefficient by an amount that will have any practical significance. It is certain that higher steam densities should increase the steam film coefficient, but the only available data show effects which can be more readily ascribed to factors on the liquor side.

Steam Velocity.

The effect of steam velocity is also obvious from theory, but without experimental confirmation. In practice the effect of changing steam velocity is often complicated by the effect of the device used on the behavior of non-condensed gases. In the horizontal type, steam velocity is relatively high and positive; in the vertical it is low and indefinite. Claassen[26] calculates the steam velocities in a vertical evaporator by assuming that the heat transfer coefficient is uniform over the whole heating surface. The velocities in the inlet nozzle, for the

[25] Ulrich, *Centr. Zuckerind.*, *32*, 218 (1924).
[26] *Z. Ver. deut. Zuckerind.*, *61*, 975-82 (1911).

evaporator he chose, were from 230 to 60 feet per second in the various bodies of a quadruple effect. By the time the steam had traversed ⅞ of the heating surface, these had dropped to 0.6-1.2 feet per second. He does not consider it possible to make the corresponding calculations for a horizontal evaporator.

The interior cross-section of a ⅞-inch 16 gage tube (used in smaller horizontal evaporators) is about 0.44 square inch. This means that a 10-inch steam pipe, with an area of 81.6 square inches, could feed 185 tubes with no decrease in velocity over that in the line. The area between two adjacent 2½-inch tubes 5 feet long, located on centers 3⅛ inches apart (fair practice in a vertical evaporator) is 37 5 square inches, or nearly half that of a 10-inch pipe. Steam entering the steam space of a vertical tube evaporator has many such paths open at its inlet Thus, some idea of the relative velocities in the two types may be formed. If equal and equally definite velocities could be maintained in the two types, and if the steam velocity in the vertical were mainly horizontal, the vertical would show the greater turbulence and, hence, possibly a higher steam film coefficient. All these factors are so interrelated with one another that no definite statements can now be made.

Steam Distribution.

Closely connected with steam velocity is steam distribution. In the horizontal it is quite certain that the long path provided in the Wellner-Jelinek design is not necessary. The steam is practically all condensed before the last passes are reached. It is questionable if 1¼-inch tubes over 8 feet long in a standard horizontal do much work in the last few feet. A good coefficient and a good temperature drop condense steam very rapidly, hence the modern tendency is to eliminate the multipass feature of the Wellner-Jelinek and feed all the tubes in parallel.

In the vertical evaporator such distribution is not so necessary because of the large areas between the tubes, even with close spacing. In building large verticals of the central downtake type the steam belt casting is often bulged to give a steam distribution space (Figure 41). Sometimes this space is closed with plates having spaces between them, the spaces being larger on the side opposite from the inlet (Figure 54). Some makers supply the belt of Figure 41 with several steam inlets. In the basket type, which usually has its steam inlet at the center of the top tube sheets, tubes may be omitted on radial lines to act as steam distribution lanes. All these devices are of doubtful value. They are based on a qualitative theory for which there are no quantitative data. Evaporators without them seem to be operating satisfactorily, though such statements from the average plant are very poor evidence. The writer knows of one vertical evaporator, provided with four inlets on the steam belt, which operated at the same capacity, so far as could be told without actual tests, when only one inlet was used. These devices are, as a rule, inexpensive, and a precaution on the safe side, but not important.

Non-condensed Gases.

The removal of non-condensed gases is a subject about which there has been endless discussion. These gases consist, in the ordinary case, of air in the heating steam, air dissolved in the feed liquor, and air entering through leaks. In the case of some solutions, other gases are evolved from the liquid as it boils (ammonia from sugar mill liquors, SO_2 from sulfite pulp liquors, etc.). In the remainder of this discussion these gases will all be referred to as air, for simplicity. Their effect on heat transfer was discussed in Chapter 6.

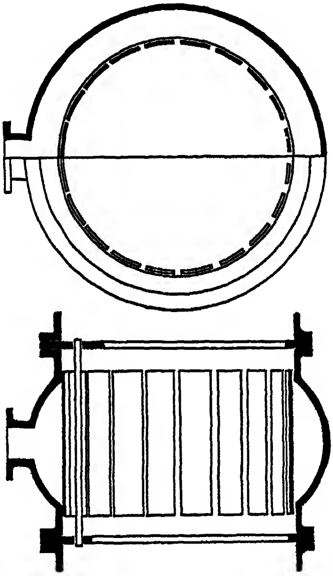

FIG. 54.—Steam distribution plates for vertical tube evaporators.

The only clue to the proper location of air vents is furnished by a consideration of steam velocity. In a horizontal evaporator the air is all swept to the exit steam chest in a positive manner, and hence steam in the tubes contains a minimum of air. On the other hand, in the vertical calandria[27] air is more or less distributed throughout the steam space, due to lack of positive velocities.[28] This same statement holds true in general for all types where steam is outside the tubes, though it may be more or less modified in special constructions. Long tubes in a narrow bundle with steam introduced at one end make possible a fairly positive steam flow. In a basket type vertical in the author's laboratory, with a basket 4 feet long, 18 inches in diameter, and a steam inlet at the center of the top tube sheet, the steam flow lines were so definitely down the basket that air vents in the top tube sheet were of no use. When they alone were open, and condensate was collected in a receiver without any air vent, in less than 20 minutes the evaporator had stopped boiling, though steam pressure and vacuum had not been changed. On opening a vent from the condensate receiver, the evaporator at once began to boil vigorously and the steam valve had to be opened several turns to maintain the pressure. Similar conditions would exist in any long, narrow steam chamber such as the Kestner or the Yaryan. Air removal from the Lillie was mentioned in the description of that machine.

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[27] A general term meaning the heating unit in a vertical evaporator of any type; sometimes used to refer to the heating unit in an evaporator with a central downtake to distinguish it from the basket type.

[28] See work by Kerr on a laboratory vertical; Bull. La. Agr. Exp. Sta. No. 149, pp. 14-21.

In vertical evaporators where the calandria is wide for its height, the lines of steam flow are nearly horizontal, and hence the air outlets should be placed around the circumference of a basket, or at the side opposite the inlet in the central downtake type. In a basket with steam lanes, they should be on the circumference halfway between the lanes. In general, the air vent is always to be placed at the ends of the lines of flow.

Unless there is a definite path for the steam so that air is driven to one point, a decision as to whether or not the air vent should be at the top or at the bottom of the steam space cannot be reached. Common practice is to vent air from both the top and the bottom of the steam space. This may be done by connecting an air vent to the top tube sheet, and using an over-size condensate pump to remove air from the bottom; or by putting a perforated pipe inside the steam space connected to the top air vent but extending nearly to the bottom of the steam space. In the case of gases lighter than steam (ammonia from sugar juices) vents at the top of the steam space are nearly sufficient. Mention should be made of the fact that wet ammonia is corrosive to brass or copper, and if there are pockets where ammonia can collect under copper tube sheets or around copper pipes or brass fittings, corrosion will be very rapid.

To direct the flow of steam in a vertical evaporator, baffles are sometimes put in the steam space. By forcing the steam to follow a regular path, air can be swept to one point and removed more positively. Such baffles were first used by Cail[29] before 1900. This construction[30] is used in the Webre evaporator (Figure 55); but most vertical evaporators are built without any such devices.

The removal of air is usually accomplished by bringing the air vent pipe outside the evaporator, and then connecting it either to the vapor space of the same body, or to the condenser. This puts the control in the hands of the operator, but there is seldom anything but "judgment" used in making this control. Kerr[31] reports that with a calandria containing baffles like Figure 55 (in a laboratory evaporator 24 inches in diameter) he was able to keep the heat transfer coefficient from falling off even with several per cent of air added to the steam, whereas this was not the case with the standard type of calandria. He suggests the control of air vents by the temperature of the mixture vented.[32] With definite steam paths and positive venting, this temperature may be nearly as high as steam temperature. It is always better to vent too much and lose a little steam than vent too little and depress the coefficient. When air is vented back into the vapor space, this causes a progressive increase from body to body of a multiple effect in the air content of the steam. The advantage of the method is that if much steam is vented along with the air, the steam is re-used in the next effect. This system is the common one. For materials evolving much non-condensable gas, such as sugar solutions, this would make the richness of the steam unduly low in the later bodies (and hence decrease the coefficient in just that part of an

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[29] Bolk, *Arch. Suikerind.*, *17*, 598 (1909).

[30] Scheinemann, U. S. Pat. 1,004,087, 1911; Webre, U. S. Pat. 1,049,425, 1913; Garrigues, 1,298,925, 1919.

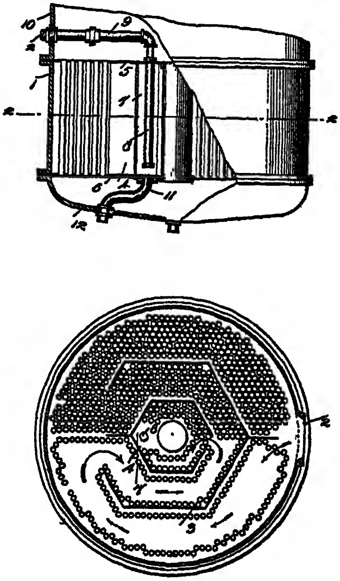

FIG. 55.—Baffles for steam space in vertical tube evaporators.

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[31] Bull. La. Agr. Exp. Sta. No. 149, pp. 14-27; *Met. Chem. Eng.*, *11*, 612 (1913).

[32] Webre, U. S. Pat. 1,049,426, 1913.

evaporator where every other factor is combining to lower it). In such cases, air should be vented directly to the condenser from each body.

Superheat.

Superheat in the steam may affect both the mean temperature drop and the heat transfer coefficient. Figure 56 is an ideal plot of the temperatures in an evaporator. Suppose the liquid throughout the evaporator is boiling at the uniform temperature T_4. Suppose the steam enters superheated to the temperature T_1 and that by the time it has traversed a distance OA along the heating surface it has cooled to the saturation temperature T_2. Over that part of the heating surface AB, it is giving up its latent heat without change in temperature. In actual practice its temperature would decrease somewhat owing to the increase in concentration of air, but this will be neglected. From B to C the heating surface is cooling the condensate from T_2 to T_3. The mean temperature drop during the transfer of superheat is Θ_s, during the transfer of latent heat Θ_L, and during cooling of the condensate Θ_c. If we knew the distances OA, AB, BC, we could calculate a weighted Θ_m, but these distances cannot even be approximated.

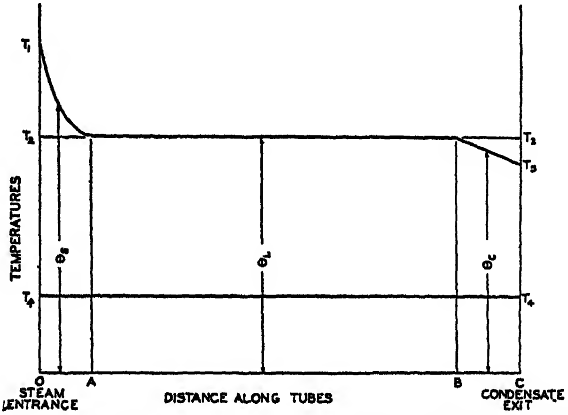

Fig 56.—Steam temperatures in horizontal tube evaporators.

Data from Chapter 4 indicate that the overall coefficient in the transfer of heat from a permanent gas to boiling water is from 1/25 to 1/1000 of the coefficient between saturated steam and boiling water. However, this will only be the case with superheated steam when the metal is dry. If the metal is wet we have the same conditions as with saturated steam. In a vertical evaporator the erratic movement of the steam probably prevents any part of the heating surface becoming dry. In the horizontal evaporator, the inlet end of the tubes may conceivably be dry with excessive superheat. The writer has carried out tests in an experimental horizontal evaporator with superheat up to 25° F., without being able to determine any drop in the coefficient as compared to saturated steam under the same conditions. Kerr[33] was unable to determine any effect of superheat up to 40° F. in an experimental vertical evaporator. Such few tests as have been reported on plant apparatus[34] show no differences (within the experimental error) for superheat up to considerably more than the above amounts. In other words, it is very unlikely that in plant apparatus any part of the heating surface is dry; consequently no part of the heating surface is apt to be working with the low coefficients usually met when transferring heat from permanent gases to water.

If this is true, and the heat transfer coefficient is substantially uniform throughout the steam space, we may consider the distances OA, AB, BC substantially proportional to the amounts of heat transmitted in each case. If H_s is the heat recovered as superheat, H_L the heat recovered as latent heat, and H_c the heat recovered from the condensate, the mean weighted temperature drop over the whole heating surface would be given by

$$\Theta_m = \frac{\Theta_s H_s + \Theta_L H_L + \Theta_c H_c}{H_s + H_L + H_c} \qquad (53)$$

Θ_s and Θ_c should be logarithmic means, but these terms are so small that the arithmetic mean is sufficiently accurate.

If values for Θ_m are calculated for any case apt to arise in practice, it will be found that H_s and H_c are so small as compared to H_L that the corresponding terms hardly affect the final result. The drop between the saturation temperature of the steam and the boiling point of the liquid, and the latent heat of the steam, are the two quantities of significance. This may easily be seen by considering that the specific heat of superheated steam at atmospheric pressure and moderate superheat, is about 0.48, while the latent heat is 970 B.t.u. per pound. Hence a superheat of 20° F. would involve a value for H_s only 1 per cent of the value of H_L. For all practical purposes the presence of moderate amounts of superheat in ordinary evaporator bodies *does not affect either the coefficient or the mean temperature drop.*

Condensate Temperature.

Condensate temperature affects the steam film coefficient by reason of its effect on the viscosity and thermal conductivity of the stagnant condensate film that is mainly responsible for the thermal resistance on the steam side of the tube. This temperature is usually intermediate between steam temperature and liquor temperature, nearer steam temperature in smaller machines and nearer liquor temperature in larger machines. This temperature is partly the reason for the variation in coefficients with variation in steam pressure. Thus the work of McAdams and Frost (Chapter 6, p 66) shows that the thermal conductivity and viscosity of the condensate film are very important factors in explaining the difference in vapor film coefficients when vapors other than steam are used. No quantitative data exist for evaluating this factor, and for a given type of machine, it cannot be varied by operator or designer.

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[33] *Met. Chem. Eng., 11,* 337 (1913); Bull. La. Agr. Exp. Sta. No. 149, pp. 38-40; *Sugar, 16,* June, p. 41 (1914).

[34] Saillard, *J. fabr. sucr.,* Aug. 1911; Jaks, *Z. Zuckerind. Böhm., 36,* 259-69 (1912); Pokorny, *Z. Zuckerind. Böhm., 38,* 218-24 (1914).

Condensate Drainage.

Condensate drainage is another factor which must have a bearing on steam film coefficients, but on which there are no data and which is not susceptible of control. It is obvious that on a vertical tube the condensate must all drain the whole length of a tube and hence the mean thickness of film on a long tube will be greater than on a shorter one. The horizontal is somewhat better in this respect, for at least the upper half of each tube should be fairly free of condensate, and the velocity of the steam helps to sweep the accumulated liquid to the discharge end. Sometimes a horizontal evaporator will be set with a pitch of an inch or two away from the steam inlet end. The Yaryan and the inclined evaporators are probably the best designs so far as condensate drainage is concerned, and the very long-tube verticals like the Kestner are probably the worst. This is recognized in a patent[35] which calls for plates across the steam space in the center of the tube bundle, to catch the condensate and divert it away from the lower part of the tubes. The Lillie uses tubes pitched toward its single tube sheet, but this makes the condensate drain in the opposite direction to steam flow. All these arguments are of questionable value; it is true that they indicate tendencies, but in the absence of quantitative data it is impossible to say whether or not such a factor as this has any value commensurate with the effect of these several designs on some other factor, such as liquor velocity.

Deposits on Tubes.

Deposits on the steam side of the tubes are not ordinarily considered important. In the light of work which will be presented under a discussion of the liquid film coefficient, it is plain that very minute deposits may have a decided effect on film coefficients. A steam film coefficient of 2000 is equivalent to the thermal resistance of a film of water at 212° F. only 0.0025 inch thick. Hence a deposit which would double that thickness would halve the film coefficient.

Such a deposit may be oil from the engines if exhaust steam is used in the evaporator, but this will obviously only occur in the first effect. If there is serious entrainment or foaming, some deposit might occur in succeeding effects, but is ordinarily washed off by the condensate. Claassen[36] says that on examination of an evaporator the oil film was found to affect only a few tubes immediately opposite the steam inlet, and is of no significance. Vrins[37] suggests nearly filling the steam space with water, putting a layer of gasoline or light oil on top of the water and allowing the mixture to drain very slowly (through a pet-cock on the trap, for instance). He also recommends boiling the steam space with soda and filling it with oil during the shutdowns in a sugar mill; but this would seem to be a poor remedy, as it would insure a film of oil on the tubes. In general, the steam side of an evaporator is rarely cleaned.[38]

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[35] Morris, U. S. Pat. 965,395, 1910.

LIQUID FILM COEFFICIENTS.

The factors which affect the liquid film seem to be the most important in governing heat transfer in evaporators. We may discuss them under the following heads:

Density of the liquid.
Thermal conductivity of the liquid.
Viscosity of the liquid.
Velocity of the liquid.
Scale or other deposits on the tubes.

The effect of *density* follows from the fact that if there is a stagnant film of liquid, its thickness must be a function of $\frac{Dv\rho}{z}$, and will therefore be less as ρ increases. The effect of *thermal conductivity* follows from the fact that if there is such a film, heat must pass through it by conduction. Both these are minor factors, we have no direct data on them, and their bearing need not be discussed further.

Viscosity and Velocity.

The subjects of viscosity and velocity are so closely connected that they must be discussed more or less together. In general, other things being equal, the effect of increasing viscosity is to decrease velocity, except in the relatively rare cases of forced circulation.

In a few cases data are available which illustrate change in velocity alone without change in viscosity, such as the data in Figures 48 and 49. In Figure 48, Curves 1 and 3 represent data taken on a vertical evaporator having tubes 2 inches in diameter and 30 inches long; Curve 2 represents the same evaporator with tubes 48 inches long.

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[36] *Z. Ver. deut. Zuckerind.*, *69*, 228-35 (1919).

[37] *Arch. Suikerind.*, *30*, 245-7 (1922); *Int. Sug. J.*, *24*, 381-2 (1922).

[38] A discussion on this subject in *Arch. Suikerind.*, *31*, Verslag 4, pp. 146-9, developed conflicting opinions and no definite conclusions.

Curves 1 and 3 differ only in the shape of the part of the evaporator below the tubes (see Figure 57). With the dished bottom the amount of liquid below the tubes was small compared to that in the tubes at moderate levels, the circulation was a maximum and Curve 1 was obtained. With the cone bottom there was a relatively large amount of dead liquid below the tubes, and its inertia slowed down the rate of circulation, giving the much flatter Curve 3. When the level was maintained high enough so that amount of liquid in circulation was large compared to the amount of dead liquid, the latter had no effect and the two curves coincide. This shows how minor differences in body design may affect circulation and through it the coefficient.

As discussed on page 113, the dotted curves in Figures 48 and 49 have been corrected for hydrostatic head and hence show the effect

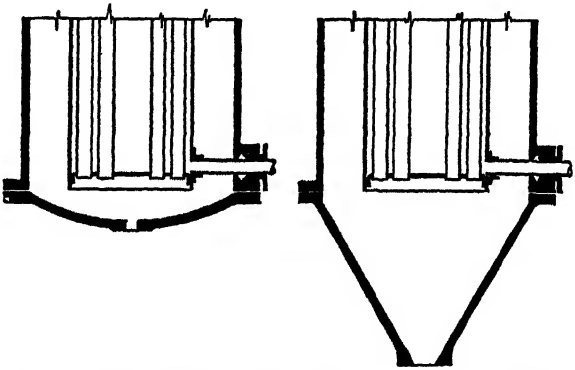

FIG. 57.—Bottoms for evaporator used in Figure 48.

of increasing velocity as the level is lowered. In these experiments the boiling point was constant and hence there was no change in viscosity. The velocity here is not known; in fact, no data are in existence at present in which velocity of natural circulation has been measured or even approximately estimated Qualitatively the curves of Figure 48 also show the effect of longer tubes on circulation. Since in these evaporators the circulation is caused by the steam evolved rising through the tube and carrying the liquid with it, it is to be expected that a longer tube, by generating a greater volume of steam, would exert a stronger pumping action. This has led designers to use longer and longer tubes, but tubes over 6 feet long are rare (except in machines of the Kestner type) because of the greater difficulty of cleaning the longer tubes. The question of tube proportions has not been systematically studied.

When one brings into the discussion a change in boiling point or a change in temperature drop, the discussion is complicated by the fact that changes in viscosity occur at the same time. Figure 58 presents data taken in the writer's laboratory.[39] These determinations were made on a vertical tube evaporator 30 inches in diameter, with a steam-basket containing twenty-four 2-inch tubes 48 inches long. The coefficients shown are apparent coefficients; i.e., calculated by using for the temperature drop the difference between steam temperature calculated from its pressure and boiling temperature calculated from the pressure of the vapor space. The original data were taken in metric units; the liquid was distilled water.

It is at once apparent that a rise in boiling point, with temperature drop constant, causes a rise in coefficient. This is due partly to the decrease in viscosity at higher temperatures, permitting stronger circulation; and partly to increased steam density and decreased condensate viscosity. At the same time an increase in temperature drop, with boiling point constant, increases the coefficient; principally because of higher velocities due to more rapid evolution of steam, and partly also because of the above mentioned changes on the steam side. In other words, most of the changes shown in these curves are functions of *liquor velocity,* partly as affected by changes in viscosity, partly as affected by changes in rate of steam evolution.

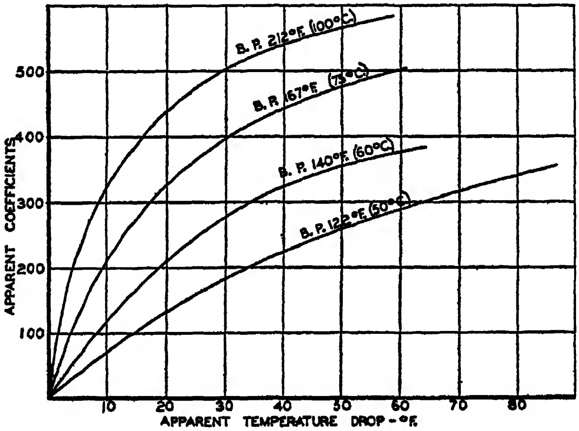

FIG. 58.—Effect of temperature drop and boiling point on heat transfer coefficients.

Walker, Lewis and McAdams[40] suggest that these changes may be expressed by assuming that $U = f\left(\frac{\Theta}{z}\right)$, and hence they plot $U \times z$ against Θ for some of the above data and some unpublished data.

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[39] The curves for 60°, 75°, and 100° C. boiling points are from Badger and Shepard, *Trans. Am. Inst. Chem. Eng., 13* (I), 101-37 (1920); *Chem. Met. Eng., 23,* 237-41, 281-4 (1920). The curve for 50° C. boiling point is from unpublished data.

[40] "Principles of Chemical Engineering," pp. 389-90.

They show that this results in a fairly straight line on logarithmic coördinates; in other words, $\Theta = a\ (Uz)^b$, where a and b are constants. Figure 59 shows the data of Figure 58 so recalculated. In addition to the data of Figure 58, Figure 59 also contains the results obtained by recalculating Curve 1 of Figure 50, which is comparable with the 140° curve of Figure 58. There is reason to believe that there was some scale in the evaporator when the data were taken for the series boiling at 212° F., and this is confirmed by Figure 59. It would seem that the vertical distance between the curves of Figure 59

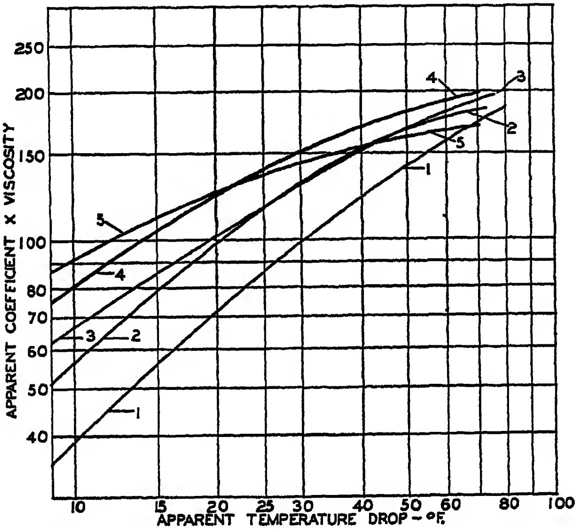


FIG. 59.—Effect of temperature drop and viscosity on apparent coefficients

Curve 1 —Vertical evaporator—boiling point 122° F
Curve 2.—Vertical evaporator—boiling point 140° F
Curve 3 —Horizontal evaporator—boiling point 140° F
Curve 4 —Vertical evaporator—boiling point 167° F
Curve 5 —Vertical evaporator—boiling point 212° F

might represent effects on the steam side, such as density of steam and viscosity of condensate (possibly air in the steam at low boiling points and low temperature drops). It should be noted that the curves run close together at temperature differences of 25° F. or more.

If the data from Figure 58 are recalculated using the true mean temperature (thus eliminating the effects of hydrostatic head), the curves of Figure 60 are obtained; and if these values for U are plotted with Θ as one coördinate and $U \times z$ as the other, Figure 61 results. Here the results for the 122° (50° C.) boiling point are seen to be relatively lower than those for 140° (60° C.), but all the curves nearly coincide. Considering that the work at 212° is almost certainly low, this would seem to indicate that for a given liquid and a given machine, correcting the coefficients for hydrostatic head and plotting in this way has eliminated differences due to viscosity and steam film conditions. If, for instance, the corrected coefficient multiplied by the viscosity is constant for any one temperature drop, irrespective of boiling point, it follows that such factors as steam density and condensate viscosity are of extremely small importance as compared to conditions on the liquor side. The hydrostatic head in the data for the horizontal tube

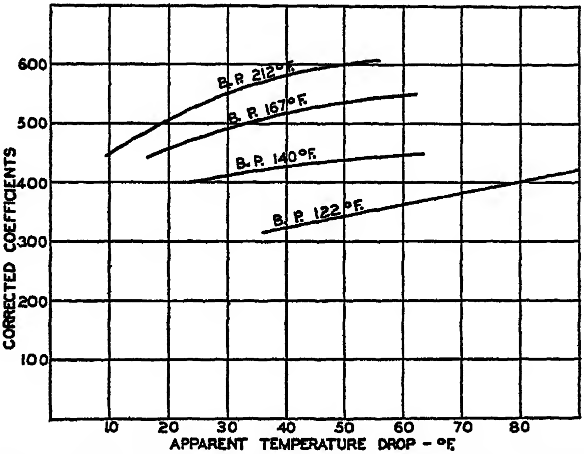


Fig. 60.—Heat transfer coefficients of Fig. 58 corrected for hydrostatic head.

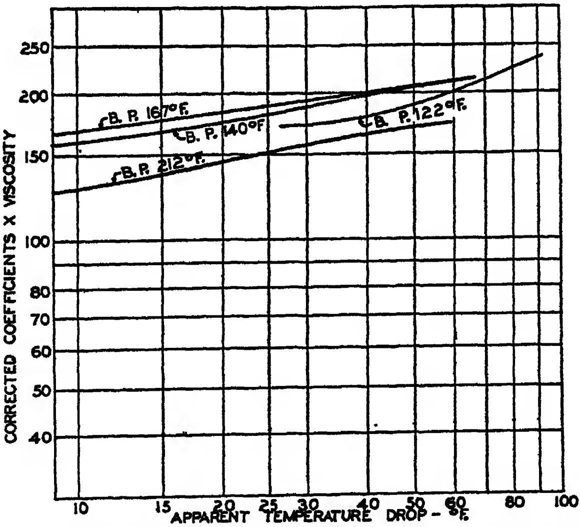


Fig. 61.—Corrected coefficients of Fig. 60 multiplied by viscosity.

evaporator given in Figure 59 is so low that the corrections make little difference. Hence the corrected curve for the horizontal has no relation to the corrected curves for the vertical evaporator.

The above discussion, together with innumerable incidents from practice, show that *velocity of the liquid is the factor that usually controls the transfer of heat from steam to boiling liquids.* This should not be considered as the only argument in all cases, as the data for the horizontal tube evaporator in Figure 59 show that at 140° the vertical and horizontal are working at about the same rate. But in general the statement is true. Viscosity is important, but merely because it affects the circulation. The data of Figure 61 may be interpreted by saying that for a given temperature drop (that is, a given rate of steam evolution) the coefficient is inversely proportional to viscosity, because with a given amount of energy available, the velocity obtained is inversely proportional to the viscosity.

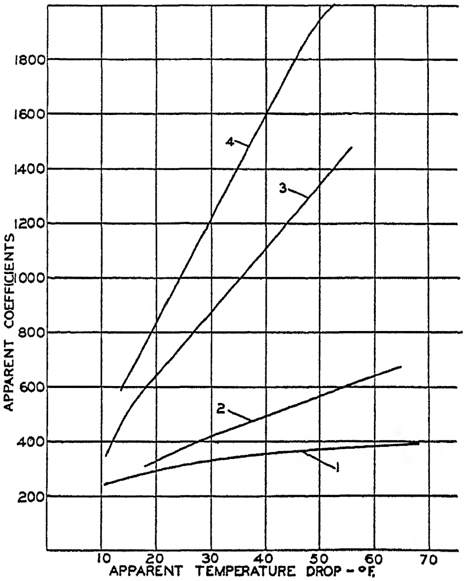


FIG. 62.—Effect of surface conditions on coefficients in horizontal tube evaporator.

Curve 1.—Rusty iron tubes.
Curve 2.—New iron tubes.
Curve 3.—Slightly dirty copper tubes.
Curve 4.—Polished copper tubes.

Effect of Surface Conditions.

The result of the presence of a layer of foreign material on the heating surface is obvious, but a better comprehension of the quantitative effect of such deposits may be had by remembering that the thermal conductivity of most scale forming materials is of the same order of magnitude as that of water. No lengthy discussion of such effects is necessary.

It is not often realized how slight a deposit is necessary to cause a noticeable decrease in the coefficient. Data reported by Pridgeon and Badger[41] on a horizontal tube evaporator illustrate this point. Figure 62 shows the range of coefficients as the tube surface was improved. Curve 1 was run with rusty iron tubes; Curve 2 with new, clean iron tubes; Curve 3 with slightly dirty copper tubes, and Curve 4 with polished copper tubes. In all cases distilled water was used, though the laboratory supply was slightly contaminated. The difference between curves 3 and 4 was caused by the removal of a film so thin that it could easily be rubbed off, leaving the copper bright, and barely making a greyish stain on the finger. A later series in the same paper showed that by treating the tubes with acid and keeping the water acid, coefficients as high as 3300 were obtained, though under these conditions the results were very erratic. Later unpublished work showed that to get comparable results the tubes had to be kept covered with distilled water at all times, scoured with scouring soap immediately before a day's run, then rinsed with water, with acid, and with water again. While a commercial machine could never hope to operate with such coefficients, this work shows the important effect of microscopic layers of scale, oxide, or dirt. Obviously, as the coefficient becomes less, the layer of foreign material would have to be thicker to make a proportionally large decrease in the coefficient. By the time the coefficient has dropped to 100, an additional sixteenth of an inch of scale would not lower it much more.

APPLICATION OF THEORY TO DESIGN.

Horizontal Evaporators.

In the horizontal tube evaporator the emphasis is all on the steam side and no especial attention is paid to the liquor side. This should not be interpreted to mean that coefficients are lower in the horizontal; a glance at Figure 48 shows how a low level increases velocity. Further, as pointed out in discussing Figure 58, under the conditions chosen for the tests a horizontal and a vertical showed practically the same capacity at the same boiling point. Kerr[42] reports a large number of tests on commercial cane sugar evaporators which are difficult to compare exactly because of questions of cleanness, etc. It should be noted, however, that his mean curves drawn to represent 21 results from vertical evaporators and 15 results from horizontal evaporators, practically coincide (Figure 63). Here Curve A shows results for the vertical type and Curve B for the horizontal type. Hence the lack

[41] *Ind. Eng. Chem., 16,* 474-8 (1924).

[42] *Trans. Am. Soc. Mech. Eng., 38,* 67-113 (1916).

of hydrostatic head in a horizontal seems to about balance the extra velocity in a vertical.

In spite of the absence of special provision for it, circulation is usually very vigorous in a horizontal evaporator. The original Wellner-Jelinek design made no provision at all for liquor circulation. In Swenson evaporators a central downtake is sometimes left, as in Figure 64. This is used only on heavy liquors or when the tube bank is deep. For very viscous materials, such as malt sirup, the tubes are located on wider centers. The Zaremba evaporator (Figure 38) depends on the sectors at the side of the tube bank for circulation, while the Buffalo horizontal (Figure 30) depends on the space between the sides of the tube bank and the flare of the body.

The writer has never seen a Buffalo horizontal in operation, but observations on Wellner-Jelineks, Swensons, and Zarembas lead to the conclusion that no device for circulation is needed for average liquids. The circulation is actually *endwise* of the tubes, the liquid boiling up vigorously at the steam inlet end, and down at the other end. This is true even with ⅞-inch tubes as short as 48 inches, and seems to be a definite result of steam density and velocity. The higher the temperature drop and the greater the rate of evaporation, the more pronounced is the difference in boiling at the two ends; doubtless because the higher rates of evaporation condense the steam faster and make more difference in steam density and velocity between the ends. In the light of this information, the long multipass steam travel of the original Wellner-Jelinek would seem necessary only for very low temperature drops and very slow evaporation. Special circulating devices may be necessary for very heavy or viscous liquids, but they are not useful with ordinary liquids in the horizontal, because they cannot be placed so as to be adapted to the natural circulation in the machine.

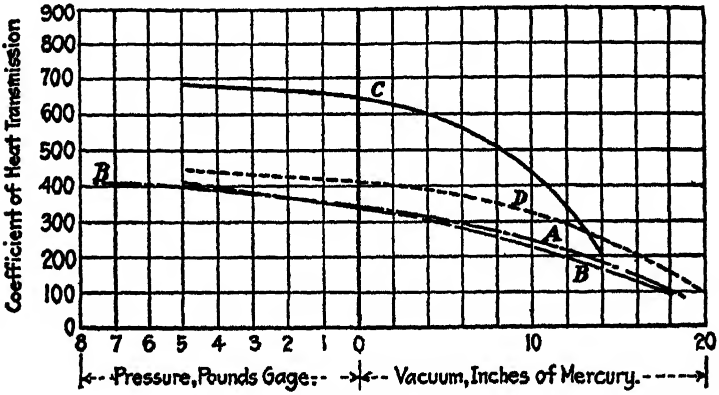


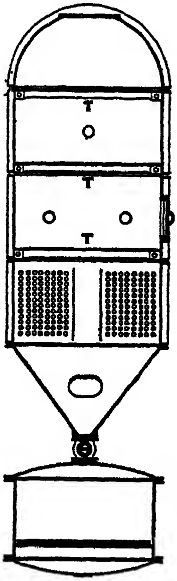
FIG. 63.—Performance of evaporators under plant conditions (Kerr).

Curve A.—Standard verticals.
Curve B—Horizontals
Curve C—Lillie
Curve D—Kestner

Vertical Evaporators.

In the vertical tube evaporator the greatest emphasis is on the liquor side, and hence it is in this type that the widest variety of devices are found to promote circulation.

The effect of changing levels is obvious from the data in Figure 48. The maximum capacity comes at the point where splashing is just sufficient to keep the tubes wet to the top. This is a rather dangerous point for operation, as slight irregularities might cause the level to fall a little below the desired point, with the result that the capacity will fall off considerably. A more important objection is that if the upper part of the tubes becomes dry through a temporary drop in level, the material being evaporated is apt to bake on and necessitate cleaning. The heavier and more viscous the liquor is, the greater danger there is in operating near the level that gives maximum capacity. Foamy liquors can be carried below the level indicated in Figure 48 without danger. Materials which tend to scale or yield crystals which adhere to the surface, usually can be carried only a little below the tops of the tubes.

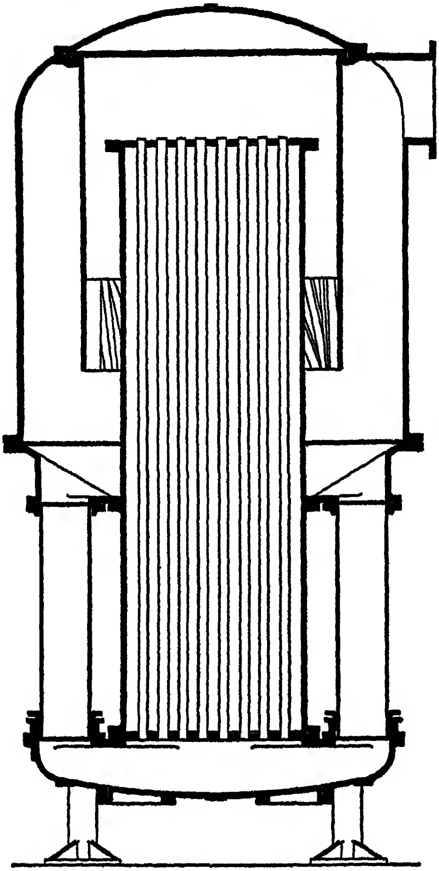


Courtesy Swenson Evaporator Co.

FIG. 64.—Horizontal evaporator with downtake.

Very early in the development of evaporators it was realized that an increase in velocity of circulation produced desirable effects. The first attempts to accomplish this took the form of hanging filler rods in the tubes, to decrease the free cross-section and hence to increase velocity[48] These devices did accomplish their purpose, but proved too troublesome to maintain, and are not now used. This idea is closely connected with the idea of *riesel* evaporators (p 114)

The bearing of tube proportions on liquor velocity may be seen from Figure 48, where, in spite of the greater hydrostatic head, a 48-inch tube is seen to give a better coefficient than a 30-inch tube. This illustrates the fallacy of qualitative reasoning in such a field, for in this case the obvious tendency of the increased hydrostatic head in a 48-inch tube (the argument used against the vertical by partisans



Courtesy Buffalo Foundry and Machine Co

FIG. 65.—Mantius high-speed evaporator.

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[48] Schwager, *Z. Ver. deut Zuckerind*, *58*, 659-64 (1908), a discussion of Ger. Pat. 335,010 is given, with lengthy calculations, by Honnicke, *Chem. App.*, *8*, 146-8, 156-8, 163-5 (1921).

of the horizontal) is more than offset by the stronger circulation, as ratio of length to diameter increases. Work in the author's laboratory, not complete enough to warrant reproducing in detail, seems to indicate that with 2-inch tubes, lengths much over 5 feet are not desirable, as then hydrostatic head more than overcomes increased velocity. This may not be true for liquids which have a tendency to foam. Further, this statement must not be taken too seriously,. as the data are limited and inaccurate. The tendency to lengthen the tubes and lower the level is carried to the extreme in the Kestner. One intermediate type of long tube high-speed evaporator is the Mantius evaporator,[44] made

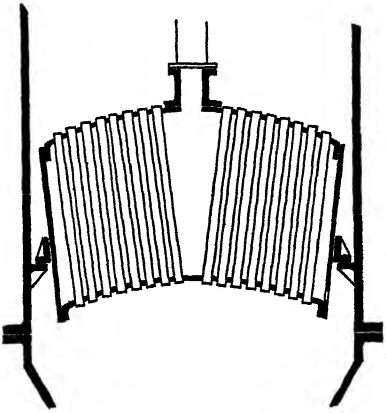

Fig. 66.—Basket type vertical with curved tube sheets.

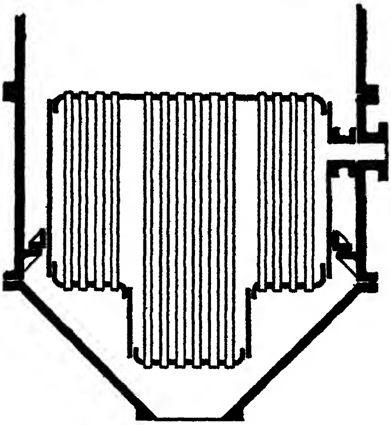

Fig. 67.—Basket type with center extension.

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[44] Mantius, U. S. Pat. 1,054,926, 1913.

by the Buffalo Foundry and Machine Company (Figure 65), in which the tubes are lengthened and the volume of liquor under the tubes is decreased.

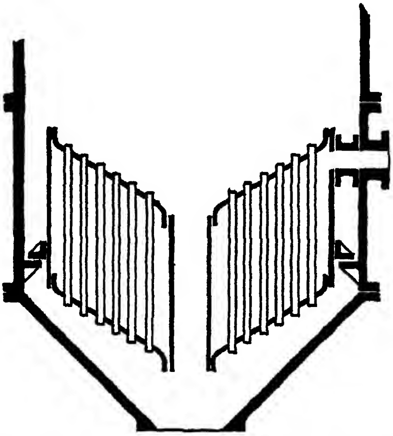

FIG. 68—Basket type with inclined tube sheets ("Express" pan).

Attempts to accelerate circulation have also led to various shapes of tube sheets. One style of tube sheet used in some basket type verticals is shown in Figure 66. The diverging angle of the tubes and the curvature of the tube sheets was considered desirable, but the costs of fabrication of this type were too great, and the advantages too speculative.

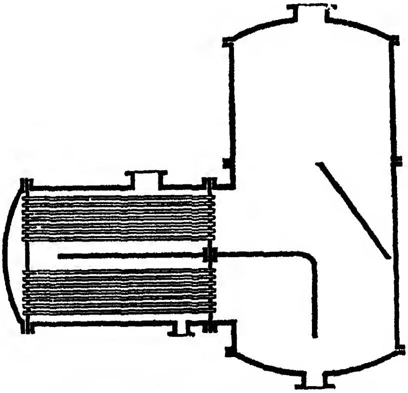

FIG 69.—Golzern-Grimma evaporator.

In cases where very viscous materials are to be boiled and where considerable material must be kept in the apparatus, as in sugar vacuum pans, special shaped tube sheets are used to insure circulation of all of the material. These shapes are such as to permit the heating surfaces to extend into the bottom of the evaporator and prevent the occurrence of pockets of stagnant material. Figures 67 and 68 show two forms, one with flat tube sheets having an extension in the middle, used with the basket type vertical; and one with inclined tube sheets, used with a central downtake, and usually known as the "express" construction.[45] The tubes may be vertical, or at right angles to the tube sheets, as shown.

Specially arranged heating surfaces may be used to promote circulation. The various arrangements of downtakes have been mentioned in Chapter 8. In some cases extra heating surface has been provided in the form of a vertical heater outside the evaporator, heated with steam of the same pressure used in the main calandria, or higher pressure. In other cases this external heater is the sole source of heat, and the evaporator body is a flash chamber only.[46] Most of these employ forced circulation, and will be dealt with under that head. One other type which may be mentioned here because the liquor is inside the tubes, though they are horizontal, is shown in Figure 69. It was originally developed by Golzern-Grimma [47] in Germany, but has been built in this country also. The division plate in the body causes the liquor to circulate from the main vessel through the lower tubes and back through the upper tubes.

Forced Circulation.

The most important developments along this line are those employing forced circulation. The earliest step was to place a propeller in the downtake. This is still the usual construction on salt evaporators. In the case of the basket type the propeller may be below the basket and driven so as to force the liquid up through the tubes. Various modifications of this idea have been suggested, one common one being to put a screw-conveyor in the downtake, especially where viscous materials are handled.

The design of propellers for such work has never been discussed. Most of the work which has been done on marine propellers is in such form as to be inapplicable. In marine work the propeller adds little to the velocity of the liquid, the velocity of approach being about the velocity of discharge. In evaporator work this is not the case. Evaporator propellers should have a pitch about equal to their diameter, and the projected area of the blades should be as large as possible. The speed should be such as to give a theoretical velocity of not over 12-15 feet per second, and the slippage at this speed may be 60 per cent.

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[45] Schroeder, *Chem. App., 2,* 118 (1915); Bendel, Germ. Pat. 323,753, 1907; *Z. Ver. deut. Zuckerind., 58,* 136 (1908); Hallström, Germ. Pat. 394,712, 1909, *ibid., 60,* 30 (1910); Forstreuter, Germ. Pat. 497,238, 1911, *ibid., 62,* 452 (1912); Engel, U. S. Pats. 1,466,357, 1923; 1,476,331, 1923; Stade, U. S. Pat. 1,168,758, 1916.

[46] Bolk, *Arch. Suikerind., 17,* 600 (1909); Fuller, U. S. Pat. 907,109, 1908; Kestner, U. S. Pat. 965,388, 1910; Naudet, U. S. Pat. 1,190,317, 1916; Reavell and Mann, U. S. Pat. 1,263,467, 1918; Barbet, U. S. Pat. 1,324,461, 1919.

[47] Schroeder, *Chem. App., 1,* 164 (1914); *2,* 102 (1915), "Die Schaumabschieder," pp. 16-27; Golzern-Grimma, Germ. Pat. 228,537, *Z. Ver. deut. Zuckerind., 60,* 1289 (1910); 357,200, 1908, *ibid., 59,* 138 (1909); 408,888, 1909, *ibid , 60,* 687 (1910); 431,214, 1909, *ibid., 60,* 1091 (1910); 443,436, 1910, *ibid., 61,* 394 (1911); 461,537, 1911, *ibid., 61,* 522 (1911).

Another means for producing velocity artificially is to inject air or any other gas under the tubes. Obviously this gas may be steam if at a pressure high enough so that it does not condense. This has long been known, though seldom used.[48] Wirth-Frey in U. S. Pat. 1,461,640, 1923, mentions it as well known. The writer has used the idea on salt evaporators very successfully. When handling very viscous materials, a slight injection of air under the tubes increases the velocity of the liquid considerably. Obviously, this is most effective at high vacua, and the use of air is impractical except in single effect evaporators or the last effect of a multiple effect. The increased capacity of vacuum pump needed is not important compared to the increase in capacity of the evaporator.

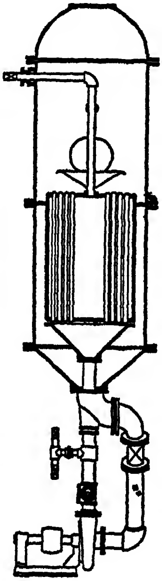

FIG. 70—Artificial circulation for vertical tube evaporators.

The most important development in this line is the idea of direct and positive pumping of the liquid through the tubes. This is most easily done in a basket type evaporator, and is covered by U. S. Pat.

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[48] Claassen, Germ. Pat. 89,784, 1895, *Z. Ver. deut. Zuckerind.*, *46*, 965-8 (1896), Steffen, Fr. Pat 278,419, 1898, *Z Ver deut. Zuckerind.*, *49*, 482 (1899); Pieper, Germ. Pats. 129,005, 129,006, 1898, *Z. Ver. deut. Zuckerind.*, *52*, 249, 414 (1902).

1,246,939, 1917[49] (Figure 70). To date this has only been used for viscous substances or substances tending to scale. Since all the discussion of factors affecting coefficients shows that those on the steam side are relatively unimportant, while those on the liquor side are of importance as they affect velocity, it would seem that this idea has not been developed nearly so far as it deserves. Experiments not yet completed indicate that very high coefficients may be obtained with moderate power inputs.

A similar idea is incorporated in the form of evaporator developed by Prache and Bouillon (Figure 71). Here liquid is forced up through a heating unit consisting of rather long, narrow tubes. From the exit of these tubes the liquid is discharged into a vaporizing chamber, from which it drops to the circulating device. This device is a special development of a screw-blade propeller. Very high coefficients are claimed for this device, though tests have not been published.

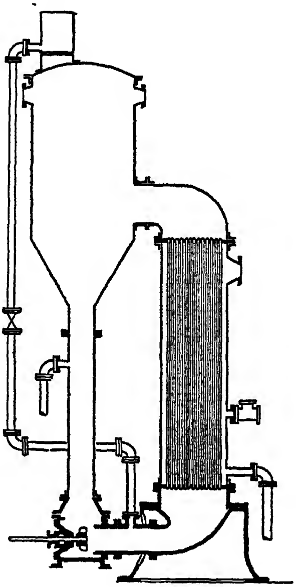

FIG. 71.—Prache and Bouillon vertical forced circulation evaporator.

Following the same line of thought, a number of evaporator constructions have been patented[50] in which the liquid is pumped through a heater in which it does not boil, and then passes to a flash vessel where evaporation takes place. The liquor is returned from this flash vessel to the circulation pump. This system differs from the Prache and Bouillon evaporator mentioned above, in that the Prache and Bouillon evaporator allows the liquid to boil in the tubes. Most of the designs involving flash only are attempts to avoid scale formation and salt deposits on the tubes They have not been used commercially to any great extent in the United States.

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[49] A partial anticipation of this idea is found in Eijdmann, U. S. Pat. 984,754. 1911.

[50] Block, U. S. Pat. 1,006,823, 1911; Prache, U. S. Pat. 1,331,373, 1920; Wirth-Frey, U. S. Pat. 1,469,475, 1923; Sebald, U. S. Pat. 1,516,314, 1924; Engl. Pats. 12,805, 1900, *J. Soc. Chem. Ind.*, *20*, 562 (1901); 158,569, 1921, *ibid.*, *41*, 620-A (1922); 208,503, 1923, *ibid.*, *43*, 573-A (1924); 214,572, 1923; Fr. Pats. 521,138, 1918, *Chim. et ind.*, *7*, 892 (1922); 543,102, 1921, *ibid.*, *10*, 279 (1923);

Inclined Evaporators.

Inclined evaporators are another logical development along the general lines under discussion. They utilize the increased velocity which a long tube tends to give, without sacrificing so much in hydrostatic head as a vertical evaporator of the same tube proportions. Van Marle[51] gives results on the evaporator shown in Figure 51, which show a rapid increase in coefficients as temperature drop increases. When plotted to take into account the change in viscosity, as was done in Figures 59 and 61, the data appear as in Figure 72. Here the effect of temperature drop in increasing circulation is apparent. Variations in Figure 72 could doubtless be partly eliminated by correcting for hydrostatic head.

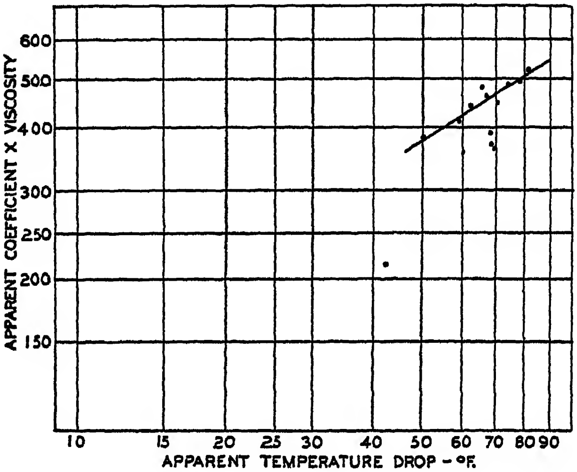

Fig. 72.—Coefficients in Buflovac inclined evaporator.

The high values obtained in these tests should not be compared directly with those from the vertical evaporator given in Figures 59

Germ. Pats. 345,257, 1918, *J. Soc Chem. Ind.*, *41*, 463-A (1922); 381,054, 1921, *ibid.*, *43*, 395-A (1924); Swed. Pat. 52,734, 1917; Forbes, *Power Plant Eng.*, *27*, 311-3 (1923).

[51] *Ind. Eng. Chem.*, *10*, 452-3 (1924).

and 61, as the vertical was fitted with iron tubes and the inclined evaporator with copper tubes. Figure 62 shows how great an influence the smoother surface of the copper tube may have had.

Film Evaporators.

With the exception of the Lillie, all film evaporators are expressly designed to take advantage of the effect of velocity on rate of heat transfer. The Kestner evaporator is one in which liquid velocity has been developed as far as natural circulation permits, by virtue of the very long tubes and low liquor level. The hydrostatic head is large, but is more or less offset by the high velocities. These velocities cannot be measured directly, but the exit velocity of the vapors as they leave the tubes can be calculated. These calculated velocities[52] vary from 45 to 170 feet per second. Obviously the liquid film cannot be travelling so fast. Direct observations show that it takes 5-20 seconds for liquid to pass through the machine, though there must be considerable error in such measurements. In spite of these excessive velocities, the figures of Depasse[53] correspond to heat transfer coefficients of 330, 230, and 145 for the three effects of a triple. In other controversies it has been indicated that the capacity of a Kestner is not much greater than that of a standard evaporator, and Reavell[54] makes no definite statements. Kerr cites six tests on Kestners and the mean curve (D in Figure 63) shows coefficients not over 10 per cent higher than the standard vertical or horizontal machines. The advantages of the Kestner undoubtedly come from the fact that its high velocity makes it possible to evaporate sensitive liquids at high temperatures (beet juices can be exposed to temperatures 10-15° F. higher than in a standard evaporator without discoloration) and also the fact that the high velocities tend to diminish scale.

The Kestner has been widely used in beet sugar mills in Europe as a pre-evaporator to increase the capacity of existing evaporators of the older types. Here its ability to work at high temperatures without discoloring the juice is a great advantage.

The Yaryan, like the Kestner, employs high velocities, but avoids the effect of hydrostatic head and has better condensate drainage. Feed distribution in a Yaryan has always been troublesome, possibly not so much so as in the Kestner. The Yaryan is subject to the same criticism as the Kestner on the score of control, as it, too, has no recirculation within the effect.

Very few tests have been published on the Yaryan. The older references cited on page 106 give some data but not in such form as to permit recalculation to coefficients. Spence[55] gives data on 14

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[52] Claassen, *Z. Ver. deut. Zuckerind.*, *61*, 980-2 (1911); Depasse, *Bull. assoc. chim. sucr. dist.*, *40*, 406-22 (1923).

[53] *Loc. cit.*—liquid being evaporated not stated.

[54] *J. Soc. Chem. Ind.*, *37*, 172-8T (1918).

[55] *Paper*, *31*, No. 26, p. 120 (Apr. 18, 1923).

Yaryans (12 triples and two quadruple effects) evaporating soda pulp liquor, in which the evaporation is expressed as average amount of water evaporated for all effects of all evaporators in each of four mills. From data given on steam and vacuum these appear to correspond to approximately 140-180 B.t.u. It cannot be stated too strongly that in such cases no comparison can be made between isolated tests under different plant conditions, due to the pronounced effect of degree of cleanness discussed on page 134.

In Figure 73 are some unpublished results of the author's on a Yaryan evaporator. In this case the liquor path was a 2½-inch iron tube 50 feet long, without return bends. The liquor space was at atmospheric pressure, the liquid was distilled water, and the steam was at 10 pounds gage. The feed was preheated to 196° F., the average boiling point was 211.0°, and the average steam temperature 239.7°.

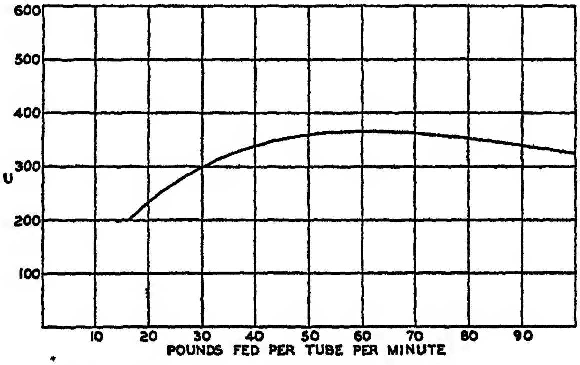

Fig 73.—Effect of rate of feed on coefficient in Yaryan evaporator.

This shows how the coefficient increased with feed till the film was fully formed, and then did not greatly decrease with higher rates of feed. With larger temperature drops, similar but flatter curves were obtained.

In such evaporators as these, high velocities can only be obtained at the cost of relatively high pressure drops. In the Kestner a high hydrostatic head is added to this friction loss. The Yaryan, though it does not have a hydrostatic head to overcome, suffers from the friction losses of the return bends. These losses have never been measured, but in the older designs were so serious as to limit the number of effects which could be operated on exhaust steam. The latest design of these bends, shown in Figure 46, was worked out to minimize this loss.

The friction losses in straight pipe due to the velocity of the steam-water mixture are shown in Figure 74. These were taken from the experimental Yaryan described above, and show the pressure drops in mm. of mercury between the feed and discharge end for different conditions, as follows:

	Average Temperature Vapor Head, ° F.	Average Temperature Steam, ° F.	Apparent Temperature Drop, ° F.
Curve 1	210.1	285.9	75.8
Curve 2	234.9	273.1	38.2
Curve 3	233.3	286.6	53.3

This shows several interesting features of film evaporators of this type. The pressure drop may be quite large; it increases with rate of feed, and with temperature drop; it decreases as the boiling point increases (due to diminished steam velocity and decreased water viscosity). The loss would, of course, be greater in a commercial machine with return bends. Such pressure drops do not have any effect on economy but do decrease capacity. The result of the increased pressure is to raise the temperature at the feed end of each coil, exactly as hydrostatic head raises the boiling point. This decreases the mean temperature drop for the whole coil, and lowers capacity. At the discharge end of the coil there is no excess pressure, any superheat in the liquid flashes into vapor, and the elevation in boiling point due to pressure is lost from the available temperature drop.

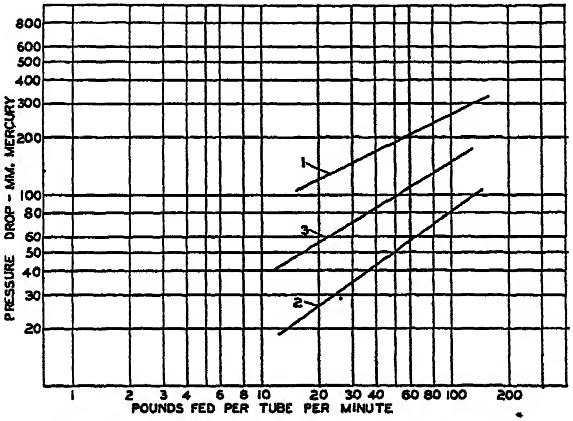

FIG. 74.—Pressure drops in Yaryan evaporator.

Where the rate of feed is high, this loss of pressure may be of the same order of magnitude as the differences in pressure between the vapor spaces of successive effects. In such a case, the available pressure drop is largely consumed in overcoming friction, leaving no temperature drop available to produce evaporation. Hence if for any reason there are large friction losses in a Yaryan, either the capacity is greatly lowered or high pressures must be carried on the first effect. This was foreseen in one of Yaryan's early patents

(383,384, 1888) in which a quadruple effect was considered. It was recommended that the first two effects be fed in parallel and that the feed then go from the first to the third effect, and from the second to the fourth. This gave the necessary pressure drop.

The Lillie evaporator is based on the idea of eliminating hydrostatic head. There is no great velocity given to the liquor, as it merely cascades over the tubes. On the steam side conditions do not seem favorable. The only published tests on Lillie evaporators are by Kerr,[56] who made four tests of quadruple effect Lillies. The average coefficients were—first effects, 570; second effects, 576, third effects, 416; fourth effects, 211. Comparing three tests on Lillie evaporators and three on Kestners in the same factory, where operating conditions were supposedly similar, the Lillie gave 17 per cent higher coefficients (Figure 63). Hence, the greater velocity of the Kestner was not enough to offset its greater hydrostatic head. As mentioned several times in discussing Kerr's tests, the effect of surface conditions is so important that much stress may not be laid on results from a limited number of factory tests under unknown conditions.

It is obviously impossible to summarize the information in this chapter and the preceding one by saying that any one type of evaporator construction is the best, or even the best for certain conditions. It appears that of all the factors affecting heat transmission, liquor velocity is so much more important than the rest, that its effect has almost totally obscured the effect of other factors in such data as are now available. In this respect the vertical would seem to have an advantage over the horizontal, but such information as we have regarding the performance of commercial machines, and a very limited amount of information from experimental work, shows no great difference between these types. Neither does published work on such high-speed evaporators as the Kestner and the Yaryan indicate any appreciable increase in capacity over the more common types.

In general, the vertical, the Kestner, the Lillie, and the Yaryan emphasize factors on the liquor side; the horizontal emphasizes conditions on the steam side. Hence, the horizontal is particularly indicated where there is no difficulty with the liquid. Viscous, foamy, scaling, or high boiling liquids call for the vertical type or one of the special evaporators. No commercial machines ever approach capacities possible in experimental apparatus, which indicates that in commercial operation more attention might well be paid to bettering surface conditions. The overwhelming importance of velocity of liquid, as shown by all theoretical and experimental work, points to the importance of forced circulation.

The choice of an evaporator for a given case calls for a consideration of so many factors, engineering and economic, that no single statement is of value. An appreciation of these many factors and their inter-relationships should form the only real guide to an engineer who must choose a machine; but even so, the quantitative data for evaluating these factors is so largely lacking that often the case seems hopeless and a good guess is the only solution. It has been the aim of these chapters to restrict this area of ignorance as far as possible.

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[56] *Trans. Am. Soc. Mech. Eng.*, *38*, 67-109 (1916), Leilich, *Met. Chem. Eng.*, *15*, 213-4 (1916), gives data from a test of an octuple effect Lillie, but gives no coefficients.

It is interesting to note the contributions of the United States to the development of evaporators. An American first fully appreciated the possibilities of multiple effect under vacuum, built the first practical multiple effect, probably suggested the vertical tube type, and is responsible for the idea of vapor heating. Although evaporators were highly developed in the German and French beet-sugar industries, the Lillie and the Yaryan are both of American origin, the only basic patent on positive artificial circulation is American, and the only systematic laboratory studies of evaporator performance that have been published were made in this country.

Chapter 10.

Theory and Calculations for Multiple Effect Evaporators.

In this discussion, it will be assumed that the reader is acquainted with the principle of multiple effect operation (the use of vapor generated in one evaporator body for heating another body boiling at a lower temperature). It is not always clear, however, as to just what the relation is between number of effects, economy, and capacity.

Economy vs. Capacity.

Consider a single effect evaporator, of S square feet heating surface, steam for heating at t_o, a boiling point of t_n under a pressure p, and a coefficient of heat transfer U. If the temperature drop between steam and liquid be $\Theta = (t_o - t_n)$, the heat transferred per hour is given by

$$Q = US\Theta$$

Now suppose there are two evaporators, both of heating surface S, the first receiving steam for heating at t_o, but boiling at some temperature t_1, which is higher than t_n. Suppose the second evaporator boils at t_n, and is heated by steam from the first at t_1. Let $\Theta_1 = t_o - t_1$ and $\Theta_2 = t_1 - t_n$. Let the coefficient of heat transfer in the two evaporators be U_1 and U_2, respectively. Now we find in practice that U is roughly the mean of U_1 and U_2. Hence, we may say that the heat transmitted in the double effect evaporator is

$$Q = U_1S\Theta_1 + U_2S\Theta_2 = US\ (\Theta_1 + \Theta_2)$$

But, this is identical with the heat transmitted in the single effect, because, by definition, $\Theta_1 + \Theta_2 = \Theta$.

In the same way, if we have a triple effect, whose mean coefficient is still U, and whose temperature drops are such that $\Theta_1 + \Theta_2 + \Theta_3 = \Theta$, we again find that the sum of the heat transmitted in three evaporators operated as a triple is the same as that transmitted through one of these evaporators operated as a single effect with the same total temperature drop. Actually this is not exactly true, but it is nearly enough true for a first approximation.

In other words, if the total temperature drop is to be the same, *a multiple effect evaporator must have as much heating surface in* *each effect as a single effect would have to produce the same evaporation.* Passing from single to multiple effect decreases capacity per square foot of total heating surface in proportion to the number of effects. On the other hand, it is obvious from the above reasoning that if one pound of steam evaporates one pound of water in a single effect, everything else being equal, it will evaporate two pounds in a double, three in a triple, and so on. Hence, the change from single to multiple effect increases *economy* but decreases *capacity.* Two evaporators, each with the same heating surface, operated as a double effect will evaporate a certain weight of water with steam temperature and vacuum held at given values. Connected as two singles, each will evaporate as much as the two together did before, but the steam consumption *per pound evaporated* will be doubled. Therefore these evaporators, operated as two single effects, will have twice the capacity, half the economy, and will condense four times as much steam as when operated as a double effect.

When the type of evaporator body for a given case has been chosen, the next problem is the number of effects which should be used. This question will be left to Chapter 14, since much of the information on which answers to it are based is developed in that chapter. Once the number and type of effects are selected, the next questions are: What should be the heating surface of each effect? At what temperatures will they operate? How much steam and water will be needed? What is the best arrangement and the optimum set of operating conditions? It is these questions that this chapter will attempt to answer, though it will appear that a satisfactory answer depends on much data seldom or never available. The most important function of this chapter will be to develop an understanding of the relationships established in a multiple effect evaporator, the nature of effects of changes in conditions, a qualitative idea of the magnitude of those changes; and, in general, to produce such a state of mind as will help the reader to approach evaporator problems intelligently. The complications which may arise in commercial installations are so numerous that only a few typical problems out of scores can be considered here.

Hausbrand's Diagrams.

The best approach to multiple effect evaporator problems is by a type of diagram extensively used by Hausbrand.[1] Figure 75 is a modified Hausbrand diagram for a straight quadruple effect evaporator. The nomenclature used is as follows:

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[1] Eugen Hausbrand (1845-1922) was for 44 years connected with the firm of C. Heckmann in Berlin. "Verdampfen, Kondensieren, und Kühlen," the most important book ever published in this field, was first published in 1899. The second German edition of 1900 was translated into English in 1903. The sixth German edition (1918) was the last.

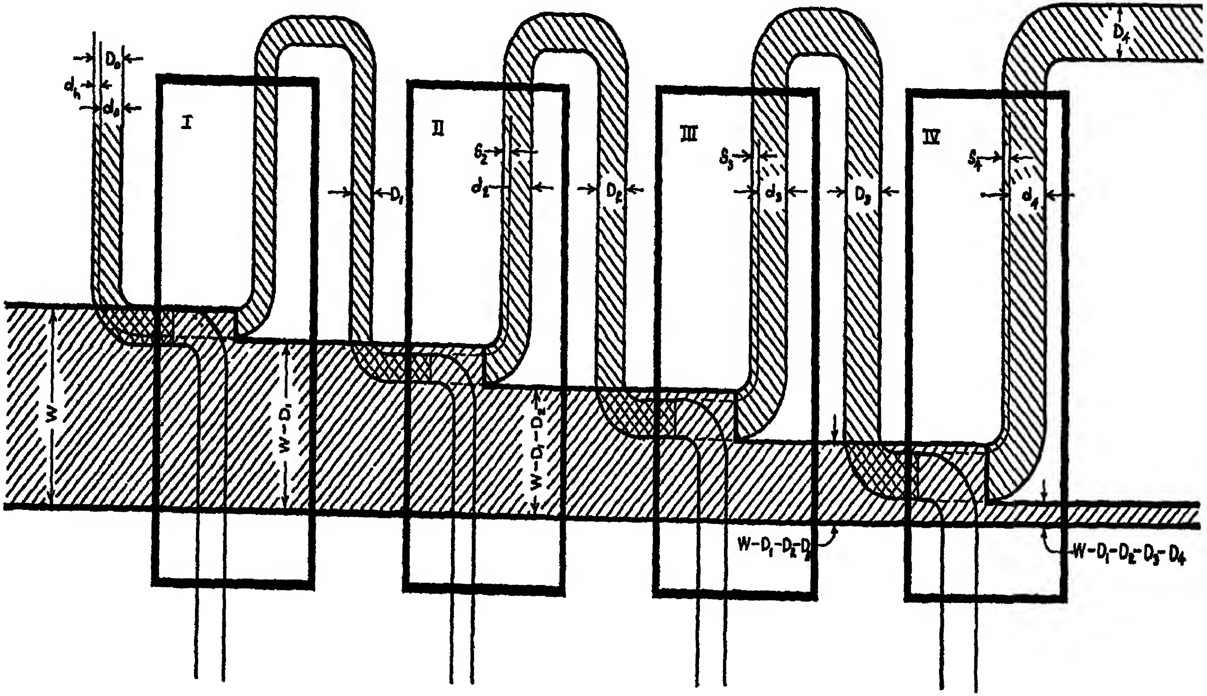
D₀
d_h
d₀
I
II
III
IV
D₁
δ₂
d₂
D₂
δ₃
d₃
D₃
δ₄
d₄
D₄
W
W−D₁
W−D₁−D₂
W−D₁−D₂−D₃
W−D₁−D₂−D₃−D₄

FIG 75—Hausbrand's diagram—forward feed

I, II, III, IV = First, second, third, and fourth effects.
W = Weight of liquid to be fed.
t_f = Temperature of feed.
t_1, t_2, t_3, t_4 = Boiling points in the four effects.
L_1, L_2, L_3, L_4 = Latent heats of vaporization in the four effects.
D_0 = Total weight of steam fed to the first effect.
L_0 = Latent heat of vaporization of steam fed to first effect.
d_h = Weight of steam needed for heating feed from t_f to t_1.
d_0 = Weight of steam needed for evaporation in first effect.
t_0 = Temperature of steam to first effect.
D_1, D_2, D_3, D_4 = Total weight of vapor leaving each effect.
d_1, d_2, d_3, d_4 = Weight of regular evaporation in each effect.
s_2, s_3, s_4 = Weight of self-evaporation in each effect.
$\Theta_1, \Theta_2\ \Theta_3, \Theta_4$ = Apparent temperature drop in each effect.
S_1, S_2, S_3, S_4 = Heating surface in each effect.
U_1, U_2, U_3, U_4 = Apparent coefficient of heat transfer in each effect.
H_1, H_2, H_3, H_4 = Total heat transmitted through the heating surface of each effect.

In Figure 75 the large rectangles represent the evaporator bodies. The cross-hatched strips represent materials, and the width of each strip represents its weight. The weight of feed entering the first effect is W, and the weight of steam entering is D_0. Part of this steam (d_h) goes to heat the feed to the boiling point, part (d_0) is used to perform evaporation. The resulting evaporation is represented by the decrease in width of the liquor line, and by the width of the line D_1, representing the weight of vapor formed. Condensate equal in weight to D_0 leaves the bottom of the first effect.

In the second effect the weight of feed is $(W - D_1)$, and the weight of steam entering is D_1. The feed, however, enters at the temperature of the first effect (t_1); but the pressure in the second effect corresponds to a lower boiling point (t_2). Therefore, the feed to the second effect is superheated and must flash. This flash is usually called *self-evaporation*. The weight of the vapor so formed is indicated by s_2. The *regular* evaporation (so called to distinguish it from the self-evaporation) is d_2, and the total weight of vapor leaving is D_2, which is in turn the sum of d_2 and s_2.

Similarly, the weight of liquid entering the third effect is $W - D_1 - D_2$, the weight of steam entering is D_2, the self-evaporation is s_3, the regular evaporation is d_3, and the total evaporation is D_3, which is $d_3 + s_3$.

For the present let us assume that we are evaporating a liquid of specific gravity = 1.0, specific heat = 1.0, no elevation of boiling point, and a viscosity the same as water. We will also assume no heat lost by radiation,[2] and that the condensate leaves each effect at steam temperature. We may then proceed to formulate each of the quantities involved as follows:

$$d_h = \frac{W(t_1 - t_f)}{L_0}$$

$$D_0 = d_0 + d_h$$

$$d_1 = \frac{d_0 L_0}{L_1}$$

$$D_1 = d_1$$

$$d_2 = \frac{d_1 L_1}{L_2} = \frac{d_0 L_0}{L_2}$$

$$s_2 = \frac{(W - D_1)\,(t_1 - t_2)}{L_2} = \frac{(W - D_1)\Theta_2}{L_2} = \frac{\left(W - \frac{d_0 L_0}{L_1}\right)\Theta_2}{L_2}$$

$$D_2 = d_2 + s_2$$

$$d_3 = \frac{D_2 L_2}{L_3} = \frac{(d_2 + s_2)L_2}{L_3} = \frac{(L_1 - \Theta_2)d_0 L_0 + \Theta_2 W L_1}{L_1 L_3}$$

$$s_3 = \frac{(W - D_1 - D_2)\Theta_3}{L_3} = \frac{(W - d_1 - d_2 - s_2)\Theta_3}{L_3}$$

$$D_3 = d_3 + s_3$$

All the quantities in the fourth effect may be similarly expressed. It is plain that all quantities involved may be expressed as functions of W, d_0, and the various L's and Θ's. This has been done as far as d_3, but the later expressions, though derived by simple substitutions of quantities already defined, become very clumsy when referred back to these initial quantities.

The difficulty with this formulation is, that we cannot predict the temperatures of the various effects, nor the weight of steam fed to the first effect. Many methods of calculation have been suggested based on assumed values for temperatures, and constant factors for the relation $D_1 : D_2 : D_3 : D_4$, but these are all arbitrary, and satisfactory only for very definite and limited conditions. It is desirable to have a general method which does not involve such fundamental arbitrary assumptions, and which may be applied to any case that may arise.

Such a method may be developed from the postulation that the *heat entering the heating surface on one side must be equal to the*

---

[2] It should be noted that the losses from ordinary apparatus are almost entirely convection, and only to a very slight extent true radiation. The term "radiation" is so universally used in this connection, however, that it has a perfectly definite meaning.

heat leaving the other side of that surface. We will begin by defining H as follows:

$$H_1 = \Theta_1 U_1 S_1,\ H_2 = \Theta_2 U_2 S_2,\ \text{etc.} \tag{54}$$

In the first effect, the heat which enters the heating surface may be expressed as $D_0 L_0$ or $(d_0 + d_h) L_0$. The heat which leaves the heating surface is $W(t_1 - t_f) + D_1 L_1$. We may then write

$$(d_0 + d_h) L_0 = H_1 = W(t_1 - t_f) + D_1 L_1 \tag{55}$$

In order to condense these formulas, square brackets around a letter denoting a weight will signify the product of that weight by its latent heat at the temperature at which it is formed or condensed, and h will be used (with a subscript denoting an effect) to represent the heat necessary to raise the liquid entering an effect to the boiling point in that effect. Equation (55) will now read

$$[d_0] + [d_h] = H_1 = h_1 + [D_1] \tag{56}$$

Similarly, for the other effects we may write

$$[D_1] = H_2 = [d_2] \tag{57}$$

$$[d_2] + [s_2] = H_3 = [d_3] \tag{58}$$

$$[d_3] + [s_3] = H_4 = [d_4] \tag{59}$$

Remembering that $D_1 = d_1$, we may write

$$\text{from (56)},\ [d_1] = H_1 - h_1 \tag{60}$$

$$\text{from (58)},\ [d_2] = H_3 - [s_2] = [d_3] - [s_2] \tag{61}$$

$$\text{and}\qquad [d_3] = [d_2] + [s_2] \tag{62}$$

Substituting the value of $[d_3]$ from equation (62) in equation (59) we have

$$[d_2] = H_4 - [s_2] - [s_3] \tag{63}$$

Combining equations (57), (60), (61), and (63), we may write

$$H_1 - h_1 = H_2 = H_3 - [s_2] = H_4 - [s_2] - [s_3] \tag{64}$$

which is the basic equation for this method of solution. If we assume a certain heating surface for each effect, and a heat transfer coefficient for each effect, the temperature drops in the various effects may then be adjusted by trial until equation (64) is satisfied.

It may not be obvious that the temperature drops in a multiple effect evaporator cannot be determined in advance. By fixing the steam pressure to the first effect and the vacuum on the last effect, the total working temperature range may be fixed, but the evaporator must come to its own equilibrium within that range so that equation (64) is satisfied. If the terms h_1, $[s_2]$, and $[s_3]$ be neglected temporarily as rather small, it will be seen that we have approximately

$$H_1 \underset{\cdot}{=} H_2 = H_3 = H_4 \tag{65}$$

which is equivalent to

$$\Theta_1 S_1 U_1 = \Theta_2 S_2 U_2 = \Theta_3 S_3 U_3 = \Theta_4 S_4 U_4 \qquad (66)$$

Since we have shown in Chapter 8 that U changes through wide limits with changes in boiling point and temperature drop, Θ must also vary from effect to effect in order to satisfy equation (66) (or rather its more exact form (64)).

Problem 1.

Conforming still to the assumptions on which this discussion has been based, let us examine a quadruple effect evaporator, fed with 1250 gallons per hour of liquid at 60° F., concentrating this to 250 gallons per hour, steam to the first effect at 5 pounds gage (227° F.) and a vacuum on the last effect of 26 inches referred to a 30-inch barometer (boiling point 125° F.). What must be the heating surface, the weight of steam used per hour, and the temperature distribution? Assume that the heating surface of all the effects is the same.

The principal difficulty in applying this or any other method of calculating evaporators, is the scarcity of data on heat transfer coefficients. The data of Figure 58 represent at present the only systematic set of experiments ever reported on a standard evaporator. Even they would require considerable and rather questionable interpolation to use by this method. It was shown, however, that if they are recalculated to corrected coefficients and plotted as in Figure 61, the variations are rather small. If we assume that the corrected coefficients of Figure 61 can be represented accurately enough for our present purposes by a single straight line, we can then reproduce a table sufficiently complete for these problems.

A single line was drawn so as to be the mean of the three upper curves of Figure 61. The points read off this were converted to apparent coefficients for different boiling points and different temperature drops, and from such data Figure 76 was drawn. It must not be forgotten that the information on which Figure 61 is based may be applied only to the design of a machine like the one in which it was determined. A change in tube proportions, liquor level, or viscosity of liquid, would invalidate this data. The most important problems in the field of evaporation are a study of such factors and their inter-relationships, so that data determined on one type of machine may be recalculated to correspond to some other type.

As illustrating the method used, consider the calculation of the apparent coefficient when the boiling point is 160° F. and the temperature drop 40° F. From the single line, UZ for a 40° drop is 195. The viscosity of water at 160° is 0.401. The pressure corresponding to 160° F. is 245.0 mm. These tests were run with a 4-foot tube, and the liquor level was at the top tube sheet, so the hydrostatic head at the bottom of the tubes was 4 feet of water.

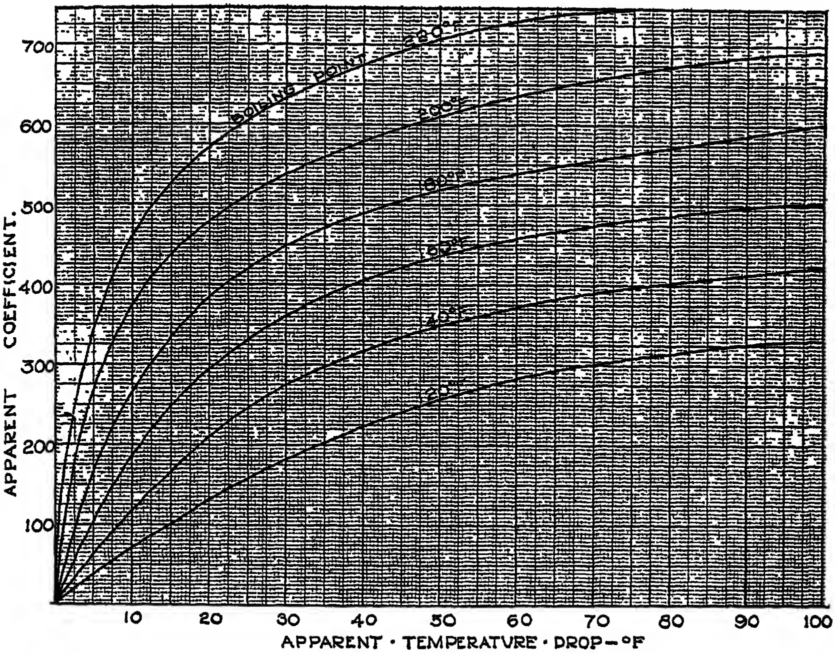

FIG. 76.—Heat transfer coefficients in vertical tube evaporator (iron tubes 2″ diam., 48″ long, boiling distilled water, liquor level even with tops of tubes).

$$4 \text{ ft. } H_2O = \frac{4 \times 12 \times 25.4}{13.6} = 89.7 \text{ mm. Hg.}$$

Pressure at top of tube—245.0 mm.
Pressure at bottom—$245.0 + 89.7 = 334.7$ mm.
Corresponding temperature at bottom—173.4° F.
Mean liquor temperature—166.7° F.
Steam temperature—200° F.
True mean temperature drop—$200° - 166.7° = 33.3°$ F.

$$\text{Corrected coefficient—}\frac{195}{0.401} = 488$$

$$\text{Apparent coefficient—}488 \times \frac{33.3}{40} = 416$$

In this way the curves of Figure 76 have been calculated. Each curve is for a constant boiling point, and gives the apparent coefficients for the apparent temperature drops indicated. Linear interpolation for intermediate boiling points is sufficiently accurate. It cannot be too often repeated that these curves hold only for a vertical evaporator with 2-inch iron tubes 4 feet long, boiling a liquid of specific gravity = 1.0, viscosity the same as water, no elevation in boiling point, with the liquor level held even with the tops of the tubes. Having this data, we may proceed to a solution of our basic equation

$$H_1 - h_1 = H_2 = H_3 - [s_2] = H_4 - [s_2] - [s_3] \qquad (64)$$

This solution must of necessity be made by the method of trial and error, the temperature drops being adjusted to satisfy equation (64) and the heating surface adjusted to produce the desired total evaporation. The feed is 10,410 pounds, the evaporation 8330 pounds, and the total temperature drop is (227° — 125°) or 102° F. In addition to satisfying equation (64) we must also have such a solution that

$$\Theta_1 + \Theta_2 + \Theta_3 + \Theta_4 = 102^\circ$$

$$\text{and} \quad d_1 + d_2 + s_2 + d_3 + s_3 + d_4 + s_4 = 8330 \text{ lbs.}$$

Having no precedent, we must first approximate the size of the heating surface. This can be done from the total heat to be transmitted, the average coefficient, and the total temperature drop. Assuming an average coefficient of about 400, an average latent heat of 990, and a first effect temperature of about 210°, we have

Heat for evaporation—8330 × 990 = 8,250,000 B t.u.
Heat for heating feed—10,410 × (210 — 60) = 1,560,000 B.t.u

Total heat to be transmitted— 9,810,000 B.t.u.

$$\text{Probable heating surface—} \frac{9{,}810{,}000}{400 \times 102} = \text{about 230 sq. ft.}$$

The next approximation is the temperature distribution. Since considerable heat must be used to heat the feed to the boiling point, the first effect temperature drop will have to be rather large. The second effect temperature drop will be small, and the third and fourth larger. For a first trial we will take at random

$$\Theta_1 = 28^\circ, \quad \Theta_2 = 20^\circ, \quad \Theta_3 = 24^\circ, \quad \Theta_4 = 30^\circ$$

With steam at 227° to the first effect, this calls for boiling points of

$$t_1 = 199^\circ, \quad t_2 = 179^\circ, \quad t_3 = 155^\circ, \quad t_4 = 125^\circ$$

Knowing the boiling point and temperature drop in each effect, we may now turn to Figure 76 and determine the coefficients, which we find to be

$$U_1 = 530, \quad U_2 = 375, \quad U_3 = 310, \quad U_4 = 205$$

The first set of calculations is as follows (asterisks mark the terms which must be equal to satisfy equation (64)):

$$H_1 = S_1\Theta_1U_1 = 230 \times 28 \times 530 = 3{,}420{,}000 \text{ B.t.u.}$$

$$h_1 = W(t_1 - t_f) = 10{,}410 \times (199 - 60) = 1{,}450{,}000 \text{ B.t.u.}$$

$$^*H_1 - h_1 = 1{,}970{,}000 \text{ B.t.u.}$$

$$d_1 = \frac{H_1 - h_1}{L_1} = \frac{1{,}970{,}000}{978} = 2020 \text{ lbs.}$$

$$W - d_1 = 8390 \text{ lbs.}$$

$$[s_2] = (W - d_1)\Theta_2 = 8390 \times 20 = 168{,}000 \text{ B.t.u.}$$

$$s_2 = \frac{(W - d_1)\Theta_2}{L_2} = \frac{168{,}000}{990} = 170 \text{ lbs.}$$

$$^*H_2 = S_2\Theta_2U_2 = 230 \times 20 \times 375 = 1{,}730{,}000 \text{ B.t.u.}$$

$$d_2 = \frac{H_2}{L_2} = \frac{1{,}730{,}000}{990} = 1750 \text{ lbs.}$$

$$H_3 = S_3\Theta_3U_3 = 230 \times 24 \times 310 = 1{,}710{,}000 \text{ B.t.u.}$$

$$W - D_1 - D_2 = W - (d_1 + d_2 + s_2) = 10{,}410 - (2020 + 1750 + 170) = 6470 \text{ lbs.}$$

$$[s_3] = (W - D_1 - D_2)\Theta_3 = 6470 \times 24 = 155{,}000 \text{ B.t.u.}$$

$$s_3 = \frac{(W - D_1 - D_2)\Theta_3}{L_3} = \frac{155{,}000}{1004} = 154 \text{ lbs.}$$

$$d_3 = \frac{H_3}{L_3} = \frac{1{,}710{,}000}{1004} = 1700 \text{ lbs.}$$

$$^*H_3 - [s_2] = 1{,}710{,}000 - 168{,}000 = 1{,}542{,}000 \text{ B.t.u.}$$

$$H_4 = S_4\Theta_4U_4 = 230 \times 30 \times 205 = 1{,}415{,}000 \text{ B.t.u.}$$

$$[s_4] = (W - D_1 - D_2 - D_3)\Theta_4 = [10{,}410 - (2020 + 1750 + 170 + 1700 + 154)] \times 30 = 4616 \times 30 = 138{,}000 \text{ B.t.u.}$$

$$s_4 = \frac{[s_4]}{L_4} = \frac{138{,}000}{1022} = 135 \text{ lbs.}$$

$$d_4 = \frac{H_4}{L_4} = \frac{1{,}415{,}000}{1022} = 1380 \text{ lbs.}$$

$$^*H_4 - [s_2] - [s_3] = 1{,}415{,}000 - 168{,}000 - 155{,}000 = 1{,}092{,}000 \text{ B.t.u.}$$

Evaporation

$d_1 = 2020$ pounds
$d_2 = 1750$
$s_2 = 170$
$d_3 = 1700$
$s_3 = 154$
$d_4 = 1380$
$s_4 = 135$
Total 7409 pounds

Substituting the terms marked with an asterisk in equation (64) we have

$$1{,}970{,}000 = 1{,}730{,}000 = 1{,}542{,}000 = 1{,}092{,}000$$

In other words, there are two faulty assumptions in our solution: (1) the total evaporation is less than desired, hence 230 square feet is too little heating surface; (2) equation (64) is not satisfied, hence the temperature distribution is not correct.

The third term in equation (64) seems about the average, hence the value assumed for Θ_3 will stand temporarily, but Θ_4 must be increased, Θ_2 decreased a little, and Θ_1 decreased considerably. The solution will be repeated using 240 sq. ft. as the heating surface and a temperature distribution of

$$\Theta_1 = 25^\circ,\quad \Theta_2 = 19^\circ,\quad \Theta_3 = 24^\circ,\quad \Theta_4 = 34^\circ$$

The results of the previous calculations are tabulated in Table XI as Trial 1. These calculations will not be repeated in detail, but the results are given as Trial 2. This gives, as a final result, for equation (64)

$$1{,}700{,}000 = 1{,}760{,}000 = 1{,}680{,}000 = 1{,}511{,}000$$

and a total evaporation of 7650 pounds instead of the desired 8330 pounds. Evidently more heating surface is needed, the temperature drop in the second effect is too high, and that in the fourth is not high enough.

Trial 3 shows the results of raising the surface to 260 square feet, cutting down Θ_2 by 1° and raising Θ_4 by 1°. This gives an evaporation of 8372 pounds—as close a check as is necessary in such calculations. The balance is

$$1{,}960{,}000 = 1{,}800{,}000 = 1{,}879{,}000 = 1{,}793{,}000$$

In Trial 4, 1° has been transferred from Θ_1 to Θ_4, with the following result:

$$1{,}820{,}000 = 1{,}820{,}000 = 1{,}906{,}000 = 1{,}838{,}000$$

Here the maximum deviation from the mean is only about 3 per cent, and a closer check cannot be obtained without splitting degrees. Obviously neither the accuracy of the data nor the uniformity of operating conditions is sufficient to warrant any closer calculations.

The average value of the terms of equation (64) for the last trial is 1,823,000. Hence, if the final adjustment of the assumptions were made so that a perfect balance was obtained, all the terms would probably have this value. Hence the first term is 3,000 B.t.u. low and the last term 35,000 B.t.u. low. Any changes of fractional parts of a degree would affect h_1, s_2, and s_3 to a very slight extent; the principal changes in making such adjustments would be in H_1 and H_4. Hence we may say that at equilibrium we would have

$$H_1 = 3{,}313{,}000 \text{ B.t.u.}$$
$$H_4 = 2{,}185{,}000 \text{ B.t.u.}$$

From this we may derive the most important information for this case:

$$D_o = \frac{H_1}{L_o} = \frac{3{,}313{,}000}{961} = 3450 \text{ lbs. steam.}$$

Heat to condenser $= H_4 + s_4 = 2{,}185{,}000 + 138{,}000 = 2{,}323{,}000$ B.t.u.

Backward Feed.

To illustrate the application of this method to another case, consider Figure 77. This is the same evaporator, but the feed comes into the fourth effect, is pumped to the third, then to the second, and the concentrated liquor is removed from the first effect. Here there is no flashing, but some of the steam entering each effect is used to heat the liquor in that effect from the temperature at which it enters, up to the boiling point. We may then write

$$H_1 = [d_1] + h_1$$
$$H_2 = [d_1] = [d_2] + h_2$$
$$H_3 = [d_2] = [d_3] + h_3$$
$$H_4 = [d_3] = [d_4] + h_4$$

$$H_4 = H_3 - h_3 = H_2 - h_2 - h_3 = H_1 - h_1 - h_2 - h_3 \qquad (67)$$

Problem 2.

What will be the effect of feeding the evaporator of Problem 1 backward? It is obvious, from an inspection of Figure 75 that the large drop across the first effect is caused by the amount of heat needed for heating feed, (h_1), and that otherwise the temperature drops would all increase from the first to the last effect. In Figure 77 the evaporation increases toward the first effect, in Figure 75 it increases toward the last effect. The magnitude of the differences is exaggerated in Figures 75 and 77. Hence in this problem we will probably have temperature drops increasing from the first to the last effect, but not increasing as fast as in Problem 1. Hence, for the first assumption, consider the same heating surface and

$$\Theta_1 = 20°, \quad \Theta_2 = 23°, \quad \Theta_3 = 26°, \quad \Theta_4 = 33°$$

This calls for

$$t_o = 227°, \quad t_1 = 207°, \quad t_2 = 184°, \quad t_3 = 158°, \quad t_4 = 125°$$

From Figure 76 we find

$$U_1 = 520, \quad U_2 = 420, \quad U_3 = 330, \quad U_4 = 220$$

TABLE XI

CALCULATIONS FOR PROBLEM I

Trial No.	S Sq. Ft.	Θ_1 °F.	Θ_2 °F.	Θ_3 °F.	Θ_4 °F.	t_0 °F.	t_1 °F.	t_2 °F.	t_3 °F.	t_4 °F.	U_1 B t.u.	U_2 B.t.u.	U_3 B.t.u.	U_4 B.t.u.
1	230	28	20	24	30	227	199	179	155	125	530	375	310	205
2	240	25	19	24	34	227	202	183	159	125	530	385	320	225
3	260	25	18	24	35	227	202	184	160	125	530	385	325	230
4	260	24	18	23	36	227	203	185	161	125	530	390	330	230

Trial No.	H_1 B.t.u.	h_1 B.t.u.	$H_1 - h_1$* B.t u.	d_1 Lbs.	H_2* B.t u.	$W - d_1$ Lbs.	$[s_2]$ B.t.u.	s_2 Lbs.	d_2 Lbs.	H_3 B.t u.
1	3,420,000	1,450,000	1,970,000	2020	1,730,000	8390	168,000	170	1750	1,710,000
2	3,180,000	1,480,000	1,700,000	1740	1,760,000	8670	165,000	167	1780	1,845,000
3	3,440,000	1,480,000	1,960,000	2010	1,800,000	8400	151,000	153	1820	2,030,000
4	3,310,000	1,490,000	1,820,000	1860	1,820,000	8550	154,000	156	1840	2,060,000

Trial No.	$H_3 - [s_2]$* B.t.u	$W - D_1 - D_2$ Lbs.	$[s_3]$ B t u.	s_3 Lbs.	d_3 Lbs.	H_4 B.t.u.	$H_4 - [s_2] - [s_3]$* B.t.u.	$W - D_1 - D_2 - D_3$ Lbs.	$[s_4]$ B.t.u.	s_4 Lbs	d_4 Lbs	Total Evaporation, Lbs.
1	1,542,000	6470	155,000	154	1700	1,415,000	1,092,000	4616	138,000	135	1380	7409
2	1,680,000	6723	162,000	162	1840	1,840,000	1,511,000	4720	161,000	158	1800	7650
3	1,871,000	6427	154,000	154	2030	2,100,000	1,793,000	4243	148,000	145	2060	8372
4	1,906,000	6554	158,000	158	1980	2,150,000	1,838,000	4336	156,000	153	2100	8327

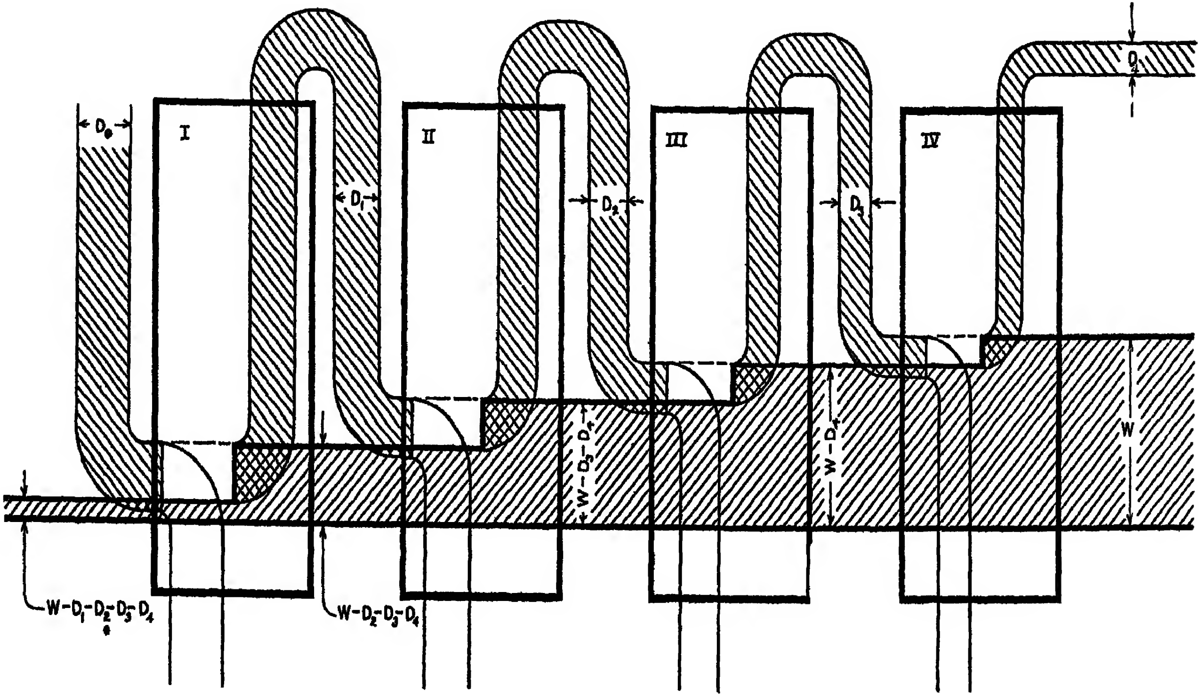
I
II
III
IV
D_e
D_1
D_2
D_3
D_4
W
$W-D_4$
$W-D_3-D_4$
$W-D_2-D_3-D_4$
$W-D_1-D_2-D_3-D_4$

FIG. 77.—Hausbrand's diagram—backward feed.

In this case it is necessary to begin the calculations with the fourth effect. Again, asterisks mark the terms which must be equal to satisfy equation (67).

$*H_4 = S_4\Theta_4U_4 = 260 \times 33 \times 220 = 1{,}890{,}000$ B t.u.
$h_4 = W(t_4 - t_f) = 10{,}410\ (125 - 60) = 677{,}000$ B.t.u.
$[d_4] = H_4 - h_4 = 1{,}890{,}000 - 677{,}000 = 1{,}213{,}000$ B.t.u.
$d_4 = \frac{[d_4]}{L_4} = \frac{1{,}213{,}000}{1022} = 1190$ lbs.
$W - d_4 = 10{,}410 - 1190 = 9220$ lbs.
$H_3 = S_3\Theta_3U_3 = 260 \times 26 \times 330 = 2{,}230{,}000$ B.t.u.
$h_3 = (W - d_4)\ (t_3 - t_4) = 9220 \times 33 = 305{,}000$ B.t.u.
$*H_3 - h_3 = 2{,}230{,}000 - 305{,}000 = 1{,}925{,}000$ B.t.u.
$d_3 = \frac{H_3 - h_3}{L_3} = \frac{1{,}925{,}000}{1003} = 1920$ lbs.
$W - d_4 - d_3 = 10{,}410 - 1190 - 1920 = 7300$ lbs.
$H_2 = S_2\Theta_2U_2 = 260 \times 23 \times 420 = 2{,}510{,}000$ B.t.u.
$h_2 = (W - d_4 - d_3)\ (t_2 - t_3) = 7300 \times 26 = 190{,}000$ B.t u.
$*H_2 - h_2 - h_3 = 2{,}510{,}000 - 305{,}000 - 190{,}000 = 2{,}015{,}000$ B.t.u.
$[d_2] = H_2 - h_2 = 2{,}510{,}000 - 190{,}000 = 2{,}320{,}000$ B.t.u.
$d_2 = \frac{[d_2]}{L_2} = \frac{2{,}320{,}000}{987} = 2{,}350$ lbs.
$W - d_4 - d_3 - d_2 = 10{,}410 - 1190 - 1920 - 2350 = 4950$ lbs.
$H_1 = S_1\Theta_1U_1 = 260 \times 20 \times 520 = 2{,}710{,}000$ B.t.u.
$h_1 = (W - d_4 - d_3 - d_2)\ (t_1 - t_2) = 4950 \times 23 = 114{,}000$ B.t.u.
$*H_1 - h_1 - h_2 - h_3 = 2{,}710{,}000 - 305{,}000 - 190{,}000 - 114{,}000 = 2{,}201{,}000$ B t.u.
$[d_1] = H_1 - h_1 = 2{,}710{,}000 - 113{,}000 = 2{,}597{,}000$ B.t.u.
$d_1 = \frac{[d_1]}{L_1} = \frac{2{,}597{,}000}{974} = 2670$ lbs.

Total evaporation:

$d_4 = 1190$ pounds
$d_3 = 1920$
$d_2 = 2350$
$d_1 = 2670$
Total 8130 pounds

Substituting terms marked with an asterisk in equation (67)

$$1{,}890{,}000 = 1{,}925{,}000 = 2{,}015{,}000 = 2{,}201{,}000$$

The first effect temperature drop is too high, the fourth effect is too small, and the evaporation is a little low. The results of repeating the calculation with 270 square feet and with

$$\Theta_1 = 19^\circ,\ \Theta_2 = 23^\circ,\ \Theta_3 = 26^\circ,\ \Theta_4 = 34^\circ$$

are given as Trial 2, Table XII. This results in an evaporation of 8502 pounds, hence 270 square feet is a little large. The balance shown by substituting in equation (67)

$$2{,}024{,}000 = 2{,}180{,}000 = 2{,}042{,}000 = 2{,}070{,}000$$

shows a maximum deviation from the average of about 5 per cent. This cannot be improved without splitting degrees. Since the calculation must be repeated for 265 square feet, this might be tried, though it is of no real significance. Trial 3, Table XII, shows the results of using 265 square feet, taking one degree off Θ_2, and dividing it between Θ_1 and Θ_3. This gives 8317 pounds total evaporation (only 13 pounds less than desired) and values for equation (67) of

$$2{,}061{,}000 = 1{,}985{,}000 = 2{,}042{,}000 = 2{,}030{,}000$$

or a maximum deviation from the mean of 2 per cent. This is an unusual check, both for evaporation and for heat balance, and few problems of this type will come out so closely. The average of the above terms is 2,029,000 B.t.u. Adjusting the last values of H_1 and H_4 to conform to this average, as was done in Problem 1, we have

$$H_1 = 2{,}628{,}000 \text{ B.t.u.}$$
$$H_4 = 2{,}029{,}000 \text{ B.t.u.}$$
$$D_0 = \frac{H_1}{L_0} = \frac{2{,}628{,}000}{961} = 2730 \text{ lbs. steam.}$$

$$\text{Heat to condenser} = H_4 - h_4 = 2{,}029{,}000 - 677{,}000 = 1{,}352{,}000 \text{ B.t.u.}$$

It might be appropriate to note at this point that the principal functions of such calculations as this is to lead one to a thorough understanding of the processes operating in a multiple effect evaporator, and the factors affecting its equilibrium, rather than an absolute determination of the quantitative results. For commercial design it is rarely desirable to carry such calculations past the first approximation or two. One soon learns to judge conditions quickly, so as to require few trials to get a fair check. Further, if the evaporation or the balance should be 10 per cent off, one should remember that in commercial operation the total temperature drop, or the feed, may easily vary by this much. Further, extra heating surface must always be provided as a safeguard against periods of irregular operation.

The greatest value of these calculations is in determining the relative magnitude of the effect of changes in arrangement or operation. For instance, comparing the results of Problems 1 and 2:

From this we may conclude that under the conditions of this problem, changing from forward feed to backward feed will (1) not change the total heating surface needed (i.e., if the evaporator is already installed for forward feed, it will still have sufficient capac-

TABLE XII

CALCULATIONS FOR PROBLEM 2

Trial No.	S Sq Ft	Θ_1 °F.	Θ_2 °F.	Θ_3 °F.	Θ_4 °F.	t_0 °F.	t_1 °F.	t_2 °F.	t_3 °F.	t_4 °F.	U_1 B.t.u.	U_2 B t u	U_3 B t.u.	U_4 B t.u.
1	260	20	23	26	33	227	207	184	158	125	520	420	330	220
2	270	19	23	26	34	227	208	185	159	125	510	430	335	225
3	265	19.5	22	26.5	34	227	207.5	185.5	159	125	515	425	335	225

Trial No.	H_4^* B.t u.	h_4 B t u.	$H_4 - h_4$ B t u	d_4 Lbs.	$W - d_4$ Lbs.	H_3 B.t.u.	h_3 B t.u.	$H_3 - h_3^*$ B t u.	d_3 Lbs.
1	1,890,000	677,000	1,213,000	1190	9220	2,230,000	305,000	1,925,000	1920
2	2,070,000	677,000	1,393,000	1360	9050	2,350,000	308,000	2,042,000	2042
3	2,030,000	677,000	1,353,000	1325	9085	2,350,000	308,000	2,042,000	2042

Trial No.	$W - d_4 - D_3$ Lbs.	H_2 B.t.u.	h_2 B.t.u	$H_2 - h_2 - h_3^*$ B.t u	$[d_2]$ B t u.	d_2 Lbs	$W - D_4 - D_3 - D_2$ Lbs.	H_1 B.t.u
1	7300	2,500,000	190,000	2,015,000	2,320,000	2350	4950	2,710,000
2	7008	2,670,000	182,000	2,180,000	2,488,000	2520	4488	2,620,000
3	7043	2,480,000	187,000	1,985,000	2,293,000	2320	4723	2,660,000

Trial No.	h_1 B.t.u.	$H_1 - h_1 - h_2 - h_3^*$ B.t u.	$[d_1]$ B.t u.	d_1 Lbs.	Total Evaporation, Lbs.
1	114,000	2,201,000	2,597,000	2670	8130
2	106,000	2,024,000	2,514,000	2580	8502
3	104,000	2,061,000	2,556,000	2630	8317

	Forward Feed (Problem 1)	Backward Feed (Problem 2)
Heating surface per effect	260	265 square feet
Steam needed $\left(\frac{H_1}{L_0}\right)$	3,450	2,730 pounds per hour
Heat to condenser..................	2,323,000	1,352,000 B.t.u. per hour
Boiling points—1st effect	203°	207°
2nd effect	185°	184°
3rd effect	161°	158°
4th effect	125°	125°

ity for backward feed); (2) decrease the steam needed by 720 pounds per hour, or 20 per cent of that needed in forward feed; (3) decrease the heat going to the condenser by 971,000 B.t.u. or 40 per cent of that in forward feed (and hence decrease the water needed for the condenser in the same proportion); and (4) will not change the working temperatures and pressures appreciably except in the first effect, which will boil at a little higher temperature with backward feed. For assistance in reaching such conclusions as these, a thorough familiarity with this method is invaluable.

Extra Steam.

One idea which we owe originally to Rillieux is that an evaporator, besides being a device for concentrating solutions, is also a very economical source of low-pressure steam. This idea is fully understood and appreciated in the sugar industry, where large volumes of steam are regularly withdrawn from various effects and used in other parts of the process. This principle and its applications are, however, very seldom appreciated outside of the sugar industry, and a few problems to illustrate their possibilities will be of interest.

Figure 78 is a development of Hausbrand's diagram in which the feed to a quadruple effect evaporator is drawn through four heaters IVa, IIIa, IIa, and Ia, on its way to the first effect. Each of these heaters is heated by steam diverted from the bodies of the evaporators. The weight of steam so diverted will be represented by E with a subscript to denote the effect from which it is withdrawn. Following Hausbrand's terminology, we will call such steam "extra steam." The analysis of this case by the balance method is as follows:

$$H_1 = h_1 + [E_1] + [d_1], \qquad [d_1] = H_1 - h_1 - [E_1] \tag{68}$$

$$H_2 = [d_1] = [d_2] + [E_2], \qquad [d_2] = H_2 - [E_2] \tag{69}$$

$$H_3 = [d_2] + [s_2] = [d_3] + [E_3], \qquad [d_3] = H_3 - [E_3] \tag{70}$$

$$H_4 = [d_3] + [s_3] = [d_4] + [E_4] \tag{71}$$

Substituting from (69) in (70)

$$H_3 = H_2 - [E_2] + [s_2] \tag{72}$$

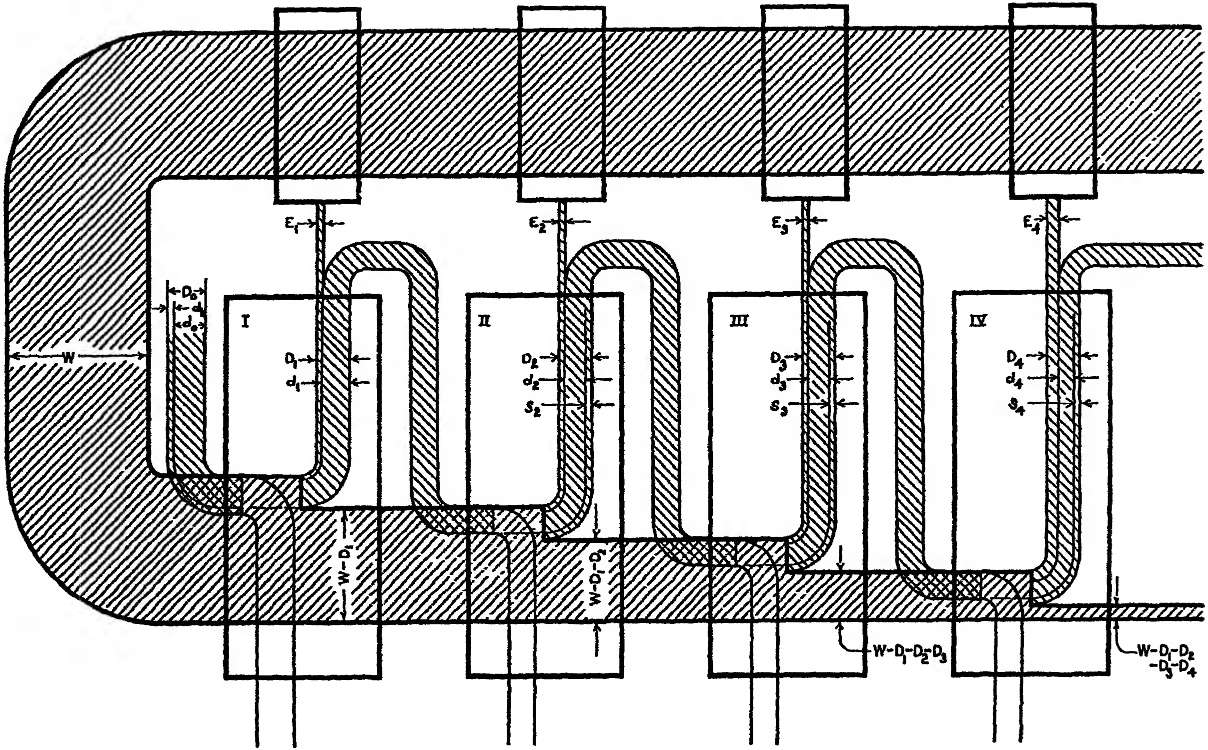
E₁
E₂
E₃
E₄
I
II
III
IV
D₀
d₀
W
D₁
d₁
D₂
d₂
S₂
D₃
d₃
S₃
D₄
d₄
S₄
W-D₁
W-D₁-D₂
W-D₁-D₂-D₃
W-D₁-D₂
-D₃-D₄

FIG 78.—Hausbrand's diagram—forward feed preheated in counter-current heaters using extra steam

Substituting from (70) in (71)

$$H_4 = H_3 - [E_3] + [s_3] \tag{73}$$

Solving (72) for H_2, substituting in (73), and solving (73) for H_2

$$H_2 = H_3 - [s_2] + [E_2] \tag{74}$$

$$H_2 = H_4 - [s_2] - [s_3] + [E_2] + [E_3] \tag{75}$$

From (68), (69), (74), and (75)

$$\begin{aligned} H_1 - h_1 - [E_1] = H_2 = H_3 - [s_2] + [E_2] = \\ H_4 - [s_2] - [s_3] + [E_2] + [E_3] \end{aligned} \tag{76}$$

which is the equation which must be satisfied by the solution in this case.

Problem 3.

Retaining all the other conditions of Problem 1, assume that the freed liquor, originally at 60° F., is fed through four vapor heaters, heated by steam from the four effects, and then passes from the first effect heater to the body of the first effect. Assume that the heaters may be so designed that the liquor is heated to within 10° F. of the temperature of the vapor used for heating.

The solution obviously calls for less heating surface in the evaporators than in the preceding cases, as a considerable amount of heat was there used for heating feed in the evaporator itself, while here all that heat passes through the surface of the external heaters. It is also obvious that the heat transmitted through the heating surface is not greatly different from effect to effect. Hence, due to the decrease in coefficients toward the last effect, the temperature drops will increase toward the last effect. Without going through the preliminary trials, we may consider the last solution in detail.

Assume a heating surface of 220 square feet per effect, and a distribution of

$$\Theta_1 = 21^\circ, \quad \Theta_2 = 21^\circ, \quad \Theta_3 = 26^\circ, \quad \Theta_4 = 34^\circ$$

From this it follows that

$$t_0 = 227^\circ, \quad t_1 = 206^\circ, \quad t_2 = 185^\circ, \quad t_3 = 159^\circ, \quad t_4 = 125^\circ$$

and from Figure 76, the coefficients are

$$U_1 = 510, \quad U_2 = 415, \quad U_3 = 335, \quad U_4 = 220$$

Since the feed is heated to within 10° of vapor temperature in each heater, the temperature of the liquid leaving each heater is

Ia — 196°, IIa — 175°, IIIa — 149°, IVa — 115°

$H_1 = S_1\Theta_1U_1 = 220 \times 21 \times 510 = 2{,}360{,}000$ B.t.u.
$h_1 = W(t_1 - 196) = 10{,}410 \times 10 = 104{,}000$ B.t.u.

$[D_1] = [E_1] + [d_1] = H_1 - h_1 = 2{,}360{,}000 - 104{,}000 = 2{,}256{,}000$ B.t.u.

$D_1 = \frac{D_1}{L_1} = \frac{2{,}256{,}000}{974} = 2310$ lbs.

$[E_1] = 10{,}410 \times (196 - 175) = 219{,}000$ B.t.u.

*$H_1 - h_1 - [E_1] = 2{,}360{,}000 - 104{,}000 - 219{,}000 = 2{,}037{,}000$ B.t.u.

$W - D_1 = 10{,}410 - 2310 = 8100$ lbs.

*$H_2 = S_2\Theta_2U_2 = 220 \times 21 \times 415 = 1{,}920{,}000$ B.t.u.

$d_2 + E_2 = \frac{H_2}{L_2} = \frac{1{,}920{,}000}{987} = 1940$ lbs.

$[E_2] = 10{,}410 \times (175 - 149) = 271{,}000$ B.t.u.

$[s_2] = (W - D_1)(t_1 - t_2) = 8100 \times 21 = 170{,}000$ B.t.u.

$s_2 = \frac{[s_2]}{L_2} = \frac{170{,}000}{987} = 172$ lbs.

$W - D_1 - D_2 = 8100 - 1940 - 172 = 5988$ lbs.

$H_3 = S_3\Theta_3U_3 = 220 \times 26 \times 335 = 1{,}920{,}000$ B.t.u.

$d_3 + E_3 = \frac{H_3}{L_3} = \frac{1{,}920{,}000}{1002} = 1920$ lbs.

$[E_3] = 10{,}410 \times (149 - 115) = 354{,}000$ B.t.u.

$[s_3] = (W - D_1 - D_2)(t_2 - t_3) = 5988 \times 26 = 156{,}000$ B.t.u.

$s_3 = \frac{[s_3]}{L_3} = \frac{156{,}000}{1002} = 156$ lbs.

$W - D_1 - D_2 - D_3 = 5988 - 1920 - 156 = 3912$ lbs.

*$H_3 - [s_2] + [E_2] = 1{,}920{,}000 - 170{,}000 + 271{,}000 = 2{,}021{,}000$ B.t.u.

$H_4 = S_4\Theta_4U_4 = 220 \times 34 \times 220 = 1{,}650{,}000$ B.t.u.

$d_4 + E_4 = \frac{H_4}{L_4} = \frac{1{,}650{,}000}{1022} = 1610$ lbs.

$[E_4] = 10{,}410 \times (115 - 60) = 572{,}000$ B.t.u.

$[s_4] = (W - D_1 - D_2 - D_3)(t_3 - t_4) = 3912 \times 34 = 133{,}000$ B.t.u.

$s_4 = \frac{[s_4]}{L_4} = \frac{133{,}000}{1022} = 130$ lbs.

*$H_4 + [E_2] + [E_3] - [s_2] - [s_3] = 1{,}650{,}000 + 271{,}000 + 354{,}000 - 170{,}000 - 156{,}000 = 1{,}949{,}000$ B.t.u.

Total evaporation:

$d_1 = 2310$ pounds
$d_2 + E_2 = 1940$
$s_2 = 172$
$d_3 + E_3 = 1920$
$s_3 = 156$
$d_4 + E_4 = 1610$
$s_4 = 130$
Total 8238 pounds

or 100 pounds less than required. One hundred pounds additional evaporation could be accomplished either by increasing the heating surface to 225 square feet or slightly increasing t_0. The balance of equation (76) is

$$2{,}037{,}000 = 1{,}920{,}000 = 2{,}021{,}000 = 1{,}949{,}000$$

This cannot be improved except by splitting degrees. The average value of the terms is 1,982,000. Hence, the first term is 55,000 B.t.u. too high, and a more exact value of H_1 would be 2,305,000 B.t.u., H_4 thus readjusted is now 1,683,000 B.t.u., from which we have

$$D_0 = \frac{H_1}{L_0} = \frac{2{,}305{,}000}{961} = 2400 \text{ pounds}$$

$$\text{Heat to condenser} = H_4 + [s_4] - [E_4] = 1{,}683{,}000 + 133{,}000 - 572{,}000 = 1{,}244{,}000 \text{ B.t.u.}$$

Comparing the results of Problems 1, 2, and 3 we have

	Problem 1 Forward Feed No Heaters	Problem 2 Backward Feed	Problem 3 Forward Feed Heaters on Extra Steam
Heating surface	260	265	225 square feet
Steam needed	3,450	2,730	2,400 pounds
Heat to condenser	2,323,000	1,352,000	1,244,000 B.t.u.
Boiling points:			
First effect	203°	207°	206°
Second effect	185°	184°	185°
Third effect	161°	158°	159°
Fourth effect	125°	125°	125°

Under the conditions of these problems, therefore, heating feed with vapor heaters saves 30 per cent of the steam used in Case I, and 12 per cent of the steam used in Case II. The evaporator may be 15 per cent smaller, but must have four external heaters. The heat going to the condenser is, of course, a direct measure of the water used in the condenser, all other conditions being equal. We find that vapor heaters save 45 per cent of the water used in Case I, and 8 per cent of that used in Case II.

It should be noted that the use of steam for heating in this manner is not greatly different from the heating of Case II. Each heater may be thought of as a part of the heating surface of the succeeding effect, built as a separate heater, partly because in this form the cost per square foot is usually less than in an evaporator, and partly because the arrangement of vapor heaters used in Problem 3 allows forward feed through the evaporator with its advantages of flash and absence of feed pumps under vacuum. Methods of feeding will be discussed fully in the next chapter; these comments are merely intended to point out the conclusions which may be reached by carrying out such calculations as here described.

The withdrawal of extra steam obviously need not be limited to heating feed. Vapor withdrawn from the first effect has already evaporated its own weight of water, and hence is just that much more economical than exhaust or live steam. Similarly, vapor withdrawn from the second effect has already evaporated twice its weight of water, and so on down the evaporator. In addition, every pound of vapor withdrawn from the evaporator to be used elsewhere, decreases the heat going from the last effect to the condenser, and hence saves water. Where water is expensive this often becomes an extremely important item. It is evident that the further down the evaporator (i.e., the nearer the condenser) the extra steam is used, the greater the saving accomplished. This is offset by the lower steam temperatures toward the last effect, which limit the uses to which such steam may be put, and increase the cost of the apparatus in which it is used, due to the lower available temperature drop as compared with live or exhaust steam. The most economical place to take off extra steam is between the last effect and the condenser. If there is any heating to be done which can be done with steam at this temperature, a heater placed in the vapor line just ahead of the condenser will usually be very economical, especially if cooling water is expensive.

To illustrate a little further the economies which may be attained by the use of extra steam, some additional examples may be of interest.

Problems 4 and 5.

In the evaporator of Problem 3, consider the effect of the withdrawal, for use in some other part of the plant, of 500 pounds of steam per hour. In Problem 4 this is to be taken from the first effect, in Problem 5, from the second. All other conditions are to be the same as in Problem 3, including the stipulation that the heating surface shall be the same in all effects. The final calculations (carried out to the nearest even degree and the nearest 10 square feet heating surface) are given in Table XIII.

It is a common idea that if such extra steam is to be removed, additional heating surface must be provided for the effect from which it is taken and for all preceding effects. The temperature distributions in Problems 4 and 5 show that no great disturbance is caused by withdrawing this amount without such extra surface; and this amount is over 20 per cent of the evaporation in the first effect, and 25 per cent of the evaporation in the second effect, of Problem 3.

To illustrate the effect of changes in heating surface, Problems 6 and 7 have been worked. Problem 6 is the same as Problem 4 but with extra heating surface in the first effect; Problem 7 is the same as Problem 5 but with extra heating surface in the first two effects.

TABLE XIII

Final Calculations for Problems 4, 5, 6, and 7

Problem	S_1	S_2	S_3	S_4	Θ_1	Θ_2	Θ_3	Θ_4	t_0	t_1	t_2	t_3	t_4	U_1	U_2	U_3	U_4
4	220	220	220	220	23	21	25	33	227	204	183	158	125	530	405	320	220
5	220	220	220	220	22	24	24	32	227	205	181	157	125	525	420	315	215
6	250	210	210	210	21	21	26	34	227	206	185	159	125	520	415	335	225
7	240	240	200	200	21	22	26	33	227	206	184	158	125	520	425	330	220

Problem	H_1	h_1	$[E_1]$	$H_1—h_1$	$*H_1—h_1—[E_1]$	D_1	$W—D_1$	$*H_2$	d_2+E_2	$[s_2]$	s_2	$[E_2]$
4	2,680,000	104,100	719,000	2,576,000	1,857,000	2640	7770	1,870,000	1900	163,000	164	260,000
5	2,540,000	104,100	250,000	2,436,000	2,186,000	2500	7910	2,220,000	2240	190,000	192	750,000
6	2,730,000	104,100	719,000	2,626,000	1,907,000	2690	7720	1,830,000	1860	162,000	164	271,000
7	2,620,000	104,000	229,000	2,516,000	2,287,000	2580	7830	2,240,000	2270	172,000	174	771,000

Problem	$W-D_1-D_2$	H_3	d_3+E_3	$[E_3]$	$[s_3]$	s_3	$*H_3+[E_2]-[s_2]$	$W-D_1-D_2-D_3$	H_4	d_4+E_4	$[s_4]$
4	5705	1,760,000	1760	343,000	143,000	143	1,853,000	3802	1,600,000	1570	125,000
5	5478	1,660,000	1660	333,000	131,000	131	2,218,000	3687	1,510,000	1470	118,000
6	5696	1,830,000	1830	354,000	148,000	148	1,939,000	3718	1,610,000	1570	127,000
7	5386	1,720,000	1720	343,000	140,000	140	2,319,000	3526	1,450,000	1420	117,000

Problem	s_4	$[E_4]$	$*H_4+[E_2]+[E_3]—[s_2]—[s_3]$	Total Evaporation
4	123	573,000	1,897,000	8301
5	115	573,000	2,270,000	8308
6	124	573,000	1,925,000	8386
7	115	573,000	2,252,000	8419

The total heating surface of the four effects is the same in all four problems.

The balances of the four problems are as follows:

Problem 4—1,857,000 = 1,870,000 = 1,853,000 = 1,897,000;
Average 1,869,000
5—2,186,000 = 2,220,000 = 2,218,000 = 2,270,000;
Average 2,224,000
6—1,907,000 = 1,830,000 = 1,939,000 = 1,925,000;
Average 1,900,000
7—2,287,000 = 2,240,000 = 2,319,000 = 2,252,000;
Average 2,274,000

Correcting H_1 and H_4 for the deviation of the last solution from the average, and assuming that in the case of Problem 3, the 500 pounds of steam is used in the process as live or exhaust steam, replaced by vapor from the evaporators in the other problems, we have

Problem	3	4	5	6	7	
Steam used in evaporator	2,400	2,790	2,680	2,830	2,710	pounds
In process	500	....		.	.. .	
Total	2,900	2,790	2,680	2,830	2,710	pounds
Heat to condenser .	1,244,000	1,122,000	1,009,000	1,136,000	1,016,000	B t u.

The differences between Problems 4 and 6, and between Problems 5 and 7, are within the error of the calculations. It is to be expected that a rearrangement of the heating surface would have no effect on economy. Taking 500 pounds steam off the first effect (Problems 4 and 6) saves 100 pounds, or 20 per cent, over using direct steam. If it is taken from the second effect, it saves 200 pounds, or 40 per cent, over direct steam. The heat going to the condenser is cut down by 110,000 B. t.u. when the extra steam is taken from the first effect, and by 230,000 B.t.u. when it is taken from the second effect.

In judging the significance of these calculations, the absolute values obtained are of small importance, the important conclusion is the relative effect of the changes in conditions. An evaporator to remove 1000 gallons water per hour is a small evaporator and would seldom be built in quadruple effect. The absolute savings in steam and water in the above problems are very small and would scarcely justify the expense of the more complicated layouts. If the evaporator were five times as large (a small beet sugar mill will have an evaporator six to eight times as large as in these problems), the saving will be of considerable importance.

General Problems.

In solving the problems so far we have confined ourselves to an ideal case. The method is adaptable to any case, however complicated. Further, in the above solutions all other conditions have been fixed and only temperature drop varied to obtain a balance. Other conditions might just as well have been taken as variables. Thus, with the total temperature drop constant, we might have considered, in any of the problems of this chapter, what the distribution of the heating surface must be to produce any fixed temperature in any or all effects. For instance, we might have demanded a uniform temperature drop in all effects, and varied the heating surface to produce it. We might have demanded a uniform heating surface in all effects, a definite temperature in some one effect, and varied the total temperature drop. Whatever conditions we impose, if we can find such a value for the variable that the balance equation is satisfied, the problem is solved, for we have actually calculated the equilibrium which will be set up in the machine under operating conditions.

No matter what the evaporator arrangement may be, a set of equations may be written and a balance equation derived in terms of the various H's. More or less than four effects, mixed feed (i.e., such order of effects as 2, 3, 4, 1, etc.), parallel feed,—in fact, any arrangement met in practice may be analyzed by this method. When one is not thoroughly familiar with the calculations, the corresponding Hausbrand diagram should be drawn and the balance equation derived. A little practice will enable one to write the balance equation directly from the diagram by inspection.

If it be desired to apply the method to a practical case, there is little greater difficulty. Radiation may be handled exactly as extra steam was handled in Problem 3. If the solution to be evaporated has a viscosity or density other than water, the curves of Figure 76 must be recalculated with due allowance for these factors. If the solution to be evaporated has an elevation in boiling point, the calculations are more tedious, especially if the solution is being concentrated, but no change in method is necessary. In such a case one must remember, as discussed on page 119, that the elevation in boiling point is lost so far as working temperature drop is concerned. That is, the first effect may boil at t_1 and its working temperature drop (Θ_1) is $t_0 - t_1$ as usual. But the vapor to the second effect is, for practical purposes, at $(t_1 - a)°$ (if a represents the elevation in boiling point of the solution at the pressure in the first effect) and hence Θ_2 is not $t_1 - t_2$, but $(t_1 - a) - t_2$.

All such cases bring in such endless complications that it is impossible to collect problems illustrating all the possibilities of the method. The reader must develop the solutions for the particular cases he has on hand, based on the principles illustrated in the problems already worked.

THERMOCOMPRESSORS.

An evaporator is, from one point of view, merely a reducing valve. We put one pound of steam into the heating surface, and get out of the vapor outlet approximately a pound of steam at a lower pressure.

Consider a single effect evaporator, with its vapor space at atmospheric pressure, the steam at 10 pounds gage, and the condensate leaving at steam temperatures. If the liquid enters the evaporator at the boiling point, it will take 970 B.t.u. to form one pound of steam. Steam at 10 pounds gage has a temperature of 238° and one pound gives up 953 B t.u. in condensing. If the condensate be cooled to 220°, the whole 970 B.t.u. needed are accounted for, and one pound of steam will evolve one pound of vapor.

The total heat of the 10 pound steam is 1160 B.t.u. To regenerate a pound of this steam from a pound of condensate at 220° would take 972 B.t.u. as a minimum, and considerably more than this if referred back to the coal pile. On the other hand, the pound of vapor contains 1150 B.t.u., and would need the addition of only 10 B.t.u. to regenerate the pound of 10 pound steam. This has given rise to the much discussed question: why not compress the vapors coming from an evaporator, use the compressed steam to heat the evaporator, and thus secure economy much greater than would be possible even with a number of effects?

Methods of Compression.

Compression of the vapors may be accomplished in two ways—by nozzles or by turbo-blowers. Reciprocating compressors were once suggested (the Piccard-Weible System[3]) but are not practical.

Nozzles are the simplest, but require the use of direct high-pressure steam to operate them. The best known is possibly the Prache and Bouillon,[4] performance tests of which are shown in Figure 79. "Entrainment ratio" is the number of pounds of vapor compressed per pound of high-pressure actuating steam. In the problem of the first paragraph of this section we considered an evaporator with vapor at 212° and a working drop of (238 — 212) or 26°. If this evaporator were operated with a nozzle compressor actuated by steam at 125 pounds gage, 0.75 pound of vapor would be compressed per pound of high-pressure steam. Thus, for every pound of steam supplied to the evaporator at 125 pounds, there would be actually 175 pounds of steam available for heating, or not quite double effect efficiency in a single body. There would still be one pound of vapor to go to other evaporators or to a condenser. On smaller temperature drops a higher compression efficiency can be obtained, and increasing the pressure of the actuating steam increases the efficiency of the nozzle. In any case, vapor corresponding to the weight of actuating steam (less losses, of course) must be disposed of elsewhere.

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[3] For a full bibliography up to 1923 see Badger, *Chem. Met. Eng.*, *28*, 27-31, 73-78 (1923).

[4] U. S. Pat. 904,276, 1908.

Turbo-blowers are possibly of more general application, due to the fact that they use power rather than steam as an actuating medium, and have higher efficiencies than a nozzle. They cannot be made to compress through very wide ranges—10 to 15 pounds is the maximum. At lower compressions they give efficiencies of 50-65 per cent (adiabatic compression being taken as 100 per cent). They have the additional advantage of not yielding excess steam to be condensed, if they are motor driven.

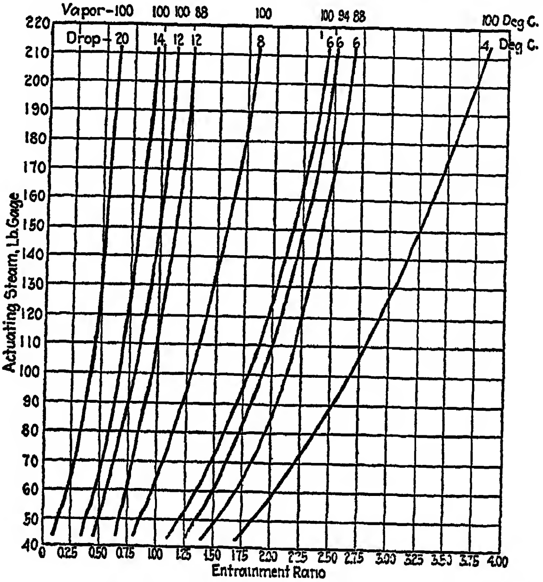

FIG. 79.—Performance of thermocompressor nozzles.

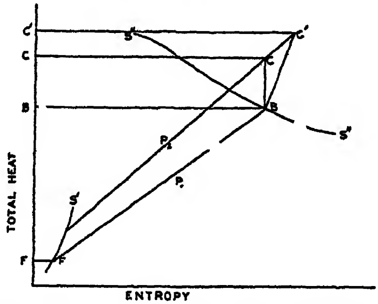

FIG. 80.—Mollier diagram for thermocompressor calculations.

Work Needed for Vapor Recompression.[5]

Figure 80 is a section of the Mollier diagram for steam. Line S'' is the curve for dry saturated steam, line S' is the condensation line, and lines P_2 and P_1 are lines of constant pressure. Let us assume that the liquor boils at P_1, the steam for heating is at P_2, the liquor enters at F, and the condensate can be cooled in heat exchangers to F.

Then to raise one pound of liquor to the boiling point and convert it to one pound of steam at P_1 takes $FB = A_1$ B.t u. If this pound of vapor were to be compressed adiabatically it would have to be compressed to C; but since, through inefficiency of the compressor more heat than the theoretical must be put into it to reach the pressure P_2, compression is represented by BC′ rather than BC. The heat available per pound of steam is therefore $FC' = A_2$. The work actually needed in kilowatts per pound of steam is therefore given by

$$W = \frac{A_2 - A_1}{3420} = \frac{BC}{3420N} \tag{77}$$

where 3420 is the factor converting B.t.u. to kilowatts, and N is the efficiency of the compressor.

If the compressor and the evaporator work as a closed system, this work of compression must be disposed of in some way. Part of it escapes as radiation, part of it as sensible heat of the condensate (if the condensate is not cooled to F), part of it is used for heating feed (for in the above discussion we have considered the feed equal to the evaporation), and part of it escapes as heat in thick liquor. If the evaporator is operating on a temperature drop which corresponds to $P_2 - P_1$, and the losses are less than W, the excess of heat supplied as work over heat lost by radiation will serve to raise P_1, decrease the working temperature drop, and hence decrease W till W is equal to the losses. Therefore, if an evaporator using thermocompression is working on a given temperature drop and it is desired to increase this drop, the only way is to increase the losses. This usually can be done most simply by blowing off steam from either the low- or the high-pressure side of the system (preferably the former). On the other hand, to decrease the working temperature drop, the losses must be lowered. In other words, the working temperature drop of a thermocompression evaporator is fixed by the losses; and such a machine will automatically adjust itself so that the losses will equal the heat input.

If the solution being evaporated has an elevation in boiling point, this increases the range through which the vapors must be compressed. If a 10° temperature drop, for instance, must be maintained, this means that the vapors must be compressed to a pressure corresponding to a temperature, not 10° higher than the temperature corresponding to the vapor space, but 10° plus the elevation in boiling point of the solution. It is important to remember that it is only the *temperature of saturation* that determines the useful working temperature drop, not the temperature of the superheated steam after compression. So, if a liquid which boils, at atmospheric pressure, at 212° is to be evaporated using a 10° temperature drop, the steam must be compressed to such a pressure that saturated steam at that pressure will have a temperature of 222° (or about 3 pounds gage). If the liquid being evaporated has a rise in boiling point of 5° F., then the vapors will have to be compressed until the temperature of saturation is 227° F., or about 4 pounds gage—an increase in power consumption in the compressor of 25 per cent.

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[5] This discussion is taken from Pridgeon, *Chem. Met. Eng.*, *28*, 1109-11 (1923).

Problem.

What would be the power consumption to evaporate 1000 pounds water per hour, at a temperature drop of 10° F., from (a) pure water, (b) a solution having an elevation in boiling point of 5° F.? In both cases evaporation is to take place at atmospheric pressure and the compressor has 60 per cent efficiency.

The total heat in one pound of vapor at atmospheric pressure and 212° F. is 1150 B.t.u. If this is compressed to give a working temperature drop of 10° F., it must be compressed to 17.9 pounds absolute. On the Mollier diagram, starting with saturated steam and atmospheric pressure (point B, Figure 80), we go straight up at constant entropy (adiabatic compression) to the line for 17.9 pounds pressure absolute. This gives us the point C, which is at 1167 B.t.u. total heat and a superheat of 23°. Hence the *theoretical* power for adiabatic compression is $1167 - 1150 = 17$ B.t.u. per pound. Equation (77) gives the power for compressing one pound of steam; multiplying by 1000 gives the total power needed for this problem:

$$\text{Power} = \frac{1000 \times 17}{3420 \times 0.60} = 8.26 \text{ kw.}$$

From the Mollier chart we can find point C′, by noting that $A_2 - A_1$ must be $\frac{17}{0.6}$ or 28.3 B.t.u. Hence the total heat at the point C′ must be $1150 + 28.3$ or 1178 B.t.u., but the pressure will still be P_2. This point is found to call for 52° superheat.

In the same way, for case (b) the steam must be compressed to 19.6 pounds absolute. BC is found, from the Mollier chart, to be 22 B.t.u. and

$$\text{Power} = \frac{1000 \times 22}{3420 \times 0.60} = 10.71 \text{ kw.}$$

with a superheat of 65°.

Capacity of Thermocompression Evaporators.

If in the above cases, it is considered that the power used for driving the compressors is furnished by condensing turbo-generators using 25 pounds steam per kw.-hour, then case (a) calls for 206 pounds steam and case (b) for 268 pounds steam, neglecting additional heat which might have to be added, over and above other losses, for such purposes as heating feed, etc. This shows a great saving over a single effect evaporator, which would take, in the above cases, at least 1100 pounds exhaust per hour, in single effect operation. Hence it would seem that a thermocompressor was very economical.

Most of the discussion in the past has been based purely on the steam economy obtainable. Too little stress has been put on the other side of the question; namely, capacity. Compare the above cases with a single effect vacuum evaporator, using exhaust steam at 5 pounds gage and boiling under a 26-inch vacuum. This gives a temperature drop for (a) of 100°, and for (b) 95°. From Figure 76, we see that for large temperature drops the coefficient at 125° F. is a little over half that at 212° F. A single effect vacuum evaporator would therefore work on ten times the temperature drop and half the coefficients of the thermocompressor, hence would need only about one-fifth the surface of the thermocompression evaporator. Thermocompression is limited by the capacity of the compressors to small temperature drops—20-30° F. at the most, and at these large drops the power is greatly increased, the economy cut down, and the superheat of the compressed steam becomes excessive. One series of patents on thermocompressors specifically limits the working temperature drop to 3° C. (5.4° F.).

The further developments of this idea have been discussed by the author,[6] but it may be said here that in a plant where exhaust steam is available for ordinary evaporators, thermocompression cannot be considered. If live steam must be used as make-up on a multiple effect evaporator having a very small drop across the first effect, the make-up steam may be added in an injector, compressing whatever vapors it will. If electrical power is very cheap and fuel high, thermocompression is the obvious method. In some cases, using high-pressure steam to operate a non-condensing turbine driving the turbo-blower, using a multiple effect evaporator designed for a small working temperature drop across the first effect, sending turbine exhaust and recompressed vapor to the first effect, and sending part of the vapors from the first effect to the compressor and the rest to succeeding effects, may work out to be the cheapest possible arrangement.

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[6] *Chem. Met. Eng.*, *28*, 27-31, 72-78 (1923).

Chapter II.

Multiple Effect Operation.

In this chapter several factors affecting the operation of multiple effect evaporators will be discussed. It is not the intention to present this as a manual for unskilled operators, but from a more or less theoretical viewpoint, for the benefit of the engineer who must decide on operating conditions.

METHODS OF FEEDING.

We may consider, as separate cases, the following general systems:

Forward feed
Backward feed
Parallel feed
Mixed feed

Forward and backward feed, as discussed in the previous chapter, mean feeding to the highest pressure or to the lowest pressure effect, respectively. The effects are always considered in the order of steam pressures, no matter what the feed order. Thus, the first effect always means the highest pressure effect; even if, as in backward feed, it is the last to receive the liquor. Parallel feed is the case where liquor is fed to all the effects in parallel, and mixed feed covers all special cases not included in the first three headings.

Forward Feed.

Forward feed is oftenest used because it is the most obvious, and is the simplest to carry out in practice. It requires but two pumps—often only one—and only one pump whose suction is under vacuum. Since in many cases in practice even the first effect boils at a pressure less than atmospheric, a feed pump is often unnecessary. Very high-pressure steam to the first effect, or a large number of effects, will sometimes put a plus pressure on the first effect, in which case a feed pump is required. In any case, thick liquor will have to be pumped out of the last effect, except in the rather rare cases where the whole evaporator operates above atmospheric pressure.

The principal consideration is, however, that in forward feed all the liquor must be heated from t_f to t_1 with single effect economy (i.e., by the direct use of steam). In turn, the liquor, once heated to t_1, passes down the effects and flashes from effect to effect. It is obvious that in an n-effect evaporator, steam formed by flash in the second effect evaporates about $(n - 1)$ times its weight, flash in the third effect evaporates $(n - 2)$ times its own weight, and so on. Although the amount of flash in any one effect is small, the total result, which is $(n - 1)s_2 + (n - 2)s_3 - - s_n$, is significant.

Another factor to be considered is the heat passing to the condenser. Since the total weight of vapor leaving each effect *increases* toward the last effect in forward feed, it would seem that this method placed an undue load on the condenser. This is true for the conditions chosen for the problems of the preceding chapter.

If, for any reason, forward feed is chosen where there is an appreciable amount of heating to do, it will generally be cheaper to do the heating in an external heater than to do it in the evaporator itself. If much heating is done in the first effect, Θ_1 must be large. This cuts down the temperature drop available for the other effects, lowers their boiling points, and thus decreases capacity both by the decrease of temperature drop available for evaporation, and by the effect of the above factors in lowering the coefficients in all later effects. This calls for practically as much more heating surface in *each* effect as is needed in the first for heating. Since the shell and piping for a heater is so much simpler than for an evaporator, a given number of square feet cost much less in a separate heater than in an evaporator body; and therefore, since n times the surface in the heater is saved in an n-effect evaporator, a heater is usually a much cheaper arrangement. Obviously, it does not affect the steam economy if the heater is fed with the same steam fed to the first effect.

If Problem 1 of the preceding chapter is repeated with feed at 190° F., we find the comparison as follows:

	Feed at 60° F.	Feed at 190° F
Heating surface per effect	260	220 square feet
Boiling points:		
First effect	203°	209°
Second effect	185°	189°
Third effect	161°	164°
Fourth effect	125°	125°

Steam used and heat to condenser are the same in both cases. The heat transfer coefficient in the heater should be about that of the first effect, hence it would need less heating surface than we have cut off the first effect, because of its larger temperature drop. The first effect is always full of liquid boiling at t_1. Even though we introduce feed at t_f, unless the volume of feed is excessive as compared with the volume of liquor in the body, there will still be maintained a temperature of t_1, and heating in the evaporator will be done with a temperature drop of Θ_1. In a heater, on the other hand, the mean temperature drop will be larger. Thus, we save in the evaporator, 160 square feet and we need add in a heater less than 40 square feet.

Backward Feed.

Backward feed is not used as often as it should be because it requires a feed pump for each effect with both suction and discharge under a vacuum. When steam pumps are used this is not a difficulty, but the chance of an air-bound centrifugal pump, due to leaky glands, is in the minds of many engineers. It is entirely possible to operate centrifugal pumps on such service without difficulty, but only with good pumps and intelligent and careful maintenance. In such evaporators as the Lillie, where there is a circulation pump for every effect in any case, backward feed causes less complications.[1]

From the problem of Chapter 10, it would appear that the major advantages of backward feed are its use of steam in multiple effect for heating and its effect on the condenser load. It is obvious that heating the feed to the temperature of the last effect is done by steam which evaporates $(n - 1)$ times its weight of water before it gets to the n^{th} effect where this heating is done. Also, the heat passing the heating surface of the n^{th} effect and used for heating feed, does not have to go to the condenser. Similarly, the heating in the next to the last effect is done by steam which first evaporates $(n - 2)$ times its weight of water, and so on. Thus, the greater the number of effects, the more economically heating is done when backward feed is used.

There are other complications which prevent the above comparisons of forward and backward feed being as general as might seem. These are mainly changes in the ratio of feed to total evaporation, changes in the properties of the solution as it is concentrated, changes in feed temperature, and the relation of the evaporator to the rest of the process.

If the feed temperature is below the boiling point in the last effect, the comparisons made above, and the qualitative conclusions drawn from the problems of Chapter 10, are unchanged. The colder the feed, the greater the advantages of backward feed. But if the solution be fed at a temperature *above* the boiling point of the last effect, all such elevation of temperature is lost in backward feed, because the liquor will flash to last effect temperature. This flash produces additional heat to go to the condenser, and the remainder of the liquid must be heated as before from the last effect temperature. As the feed temperature approaches that of the first effect, the magnitude of these changes is increased. On the other hand, the nearer the feed temperature approaches that of the first effect, the less steam is used for heating in forward feed, the more plainly marked becomes the multiple utilization of flash in the terms s_2, s_3, etc. By repeating the problems of Chapter 10, we find that after the feed gets a little above the last effect temperature, forward feed uses less steam than backward feed, and the heat going to the condenser in forward feed is less than in backward feed.[2]

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[1] Lillie, U. S. Pat. 521,215, 1894.

As the ratio of feed to evaporation becomes larger, with the feed temperature constant, the heat used for heating becomes larger in proportion to the heat used for evaporation, and hence the advantage of backward feed is somewhat more marked. At the same time, the evaporation from flash in forward feed is also larger, and a qualitative comparison is not simple. In general, larger ratios of feed to evaporation (i e., larger volumes of thick liquor) slightly increase the feed temperature up to which backward feed is more economical than forward feed.

Another factor of considerable importance is the relation of the evaporator to the rest of the process; i.e., the operation to which the thick liquor is subjected after leaving the evaporator. In forward feed the liquor leaves the evaporator at the temperature of the last effect; in backward feed, at first effect temperature. If the liquid is to be sent from the evaporator to a step in the process where it must be heated to a temperature above t_n, the steam necessary to do this heating must be figured and charged to the evaporator in the case of forward feed. If the liquid is not to be heated after evaporation, the extra temperature of the thick liquor in backward feed may constitute a loss. This consideration may reverse a conclusion based on previous considerations alone.

If the liquid which is being concentrated increases in viscosity, density, or elevation of boiling point as it concentrates, these properties have an effect on the operation of the evaporator. It was shown in Chapter 9 that the effect of hydrostatic head is exaggerated at lower pressures, hence it is desirable on this score to keep the densest liquor in the hottest effects (i.e., backward feed). Since viscosity changes very rapidly with temperature, keeping the more viscous material at a higher temperature partly offsets the increase of viscosity with concentration—again an argument for backward feed. Elevation in boiling point does not change greatly with pressure (the Dühring lines for most solutions have a slope very little greater than 1) but does decrease slightly at lower pressures. This is an argument for forward feed; but a stronger one is, that if the material is fed forward, so that the largest elevation in boiling point is in the last effect, this increases the temperature of the liquid in the later effects and thus gives a higher coefficient. With high-molecular-weight organic materials (paper liquors, packing-house tankage, etc.) where the viscosity increases greatly and the elevation in boiling point very slightly as concentration increases, the arguments for backward feed are the strongest.[3] With inorganic materials where viscosity increases more slowly and elevation in boiling point more rapidly as concentration increases, the arguments for forward feed have greater weight.

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[2] Webre, *Chem. Met. Eng*, *27*, 1073-8 (1922) has worked a number of problems by a method essentially like the author's, for a fixed ratio of feed to evaporation.

One other consideration which affects the choice of feeding methods is the effect of temperature on the material being evaporated. Sugar solutions, for instance, when dilute may be subjected to temperatures up to 240-250° F. for shorter or longer times without danger; but when concentrated, the temperatures above 185° may cause discoloration. Hence sugar solutions must always be evaporated with forward feed, so as to subject the concentrated material to the lowest temperatures. This consideration outweighs the effect of increase in viscosity and density with concentration on a choice of feeding methods. Some materials cannot be subjected to high temperatures at any concentration; these may be evaporated in single effect only. Such liquors as waste paper-liquors, tankage, etc., where the qualities of the concentrated material are not a problem, may be fed backward without harm.

In cases where a material is sensitive to heat at high concentrations, heating by vapor heaters and then feeding forward, as in Problems 3 to 7, is obviously indicated. This protects the liquid from damage, but gives the economy of backward feed without the pump difficulties of backward feed. Unfortunately, too many cases in practice are decided on the basis of the easiest way (forward feed), without consideration of the possibilities of backward feed.

Parallel Feed.

Parallel feed is a special case not often met. The only case where it is regularly used is in the salt industry, where a fully saturated brine is evaporated. The problems worked by Webre[4] show that, under the conditions assumed by him, for a small range of feed temperatures intermediate between those of the first and last effects this method of feed is slightly more economical than either forward or backward feed. The saving of steam is not great. With feed temperatures in the upper part of the range where parallel feed uses the least steam, there is also a slightly smaller amount of heat going to the condenser. These conclusions are drawn from one set of problems, and variations in the conditions would change them. The method has such limited applications (it would never be used when a solution is to be concentrated) and such doubtful savings that it is not necessary to discuss it further.

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[3] Moore, in U. S. Pat. 1,098,825, 1914, covers backward feed for caustic soda.

[4] *Loc. cit.*

Heating with Vapor Heaters.

Heating with vapor heaters as mentioned in the previous chapter, comes as a special case of backward feed (so far as steam used for heating is concerned). Obviously, if the feed is at a temperature higher than the boiling point in the last effect, one or more heaters may be omitted. According to the number of heaters used this case is intermediate between forward and backward feed; being more economical than backward feed if a heater is used on every effect, and becoming identical with forward feed if the feed temperature approximates that of the first effect. In every case the writer has investigated, it is the most economical method of all, except where there is a relatively large volume of thick liquor which must be reheated to temperatures above the first effect temperature for the next step in the process. This is true both as regards steam consumption and heat to the condenser.

This method of preheating feed is described in certain patents[5] but these patents are limited to placing the heating coils in the steam space of the evaporator body instead of in or adjacent to the vapor pipes. It has also been suggested[6] that jet condensers might be used for such heating. In this case, the condenser is located in the vapor line and juice is pumped through it, falling into a barometric leg-pipe. The disadvantage of this method is that the juice is considerably diluted; though even so the evaporation is more economical than straight forward feed. The only possible advantage of such a system would be low first cost of the condensers as compared to tubular heaters.

Mechanically, heating with vapor heaters is the same as forward feed, in that it does not require pumps under vacuum except for the thick-liquor pump. The cost of the heaters and the extra piping is a serious item in small installations; but on the other hand, the smaller the installation, the fewer the effects (usually) and hence the simpler the lay-out. There seems to be an idea that if vapors are so withdrawn, the heating surfaces of all effects may not be alike, which obviously would increase the first cost of the evaporators. The problems of Chapter 10 showed that this is not true, and that even when considerable amounts of extra steam are withdrawn, the evaporator bodies may still be all uniform.

Mixed Feed.

In special cases it may be advantageous to feed the bodies in irregular order. Consider a liquor coming to the evaporator at a high feed temperature, but to be concentrated to a very viscous and dense thick liquor, yet not sensitive to high temperatures. Caustic soda or waste paper-liquors are examples. The high feed temperature indicates forward feed, the high viscosity of the finished material indicates backward feed. The first involves a sacrifice of capacity; the second, of economy.

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[5] Lillie, U. S. Pat. 972,880, 1910; Newhall, U. S. Pat. 1,318,793, 1919; Robertson and Ballingall, Engl. Pat. 15,698, 1890.

[6] Van Moll, *Arch. Suikerind.*, *32*, 673-83 (1924); Lillie, U. S. Pat. 422,234, 1890.

The solution of this problem is indicated by the fact that the removal of equal increments of water does not increase the per cent concentration by equal increments. The last effect usually shows the major part of the total change in concentration accomplished in the whole evaporator. Thus, if a solution is to be evaporated from, say, 10 per cent to 50 per cent solids in a quadruple effect, removing equal amounts of water in each effect (sufficiently near actual conditions for this purpose) results in concentrations leaving each effect of 12.5, 16.7, 25.0, and 50 per cent respectively. Hence the principal increase in the undesirable properties will occur in the last effect. If the solution is fed into the second effect and then forward to the fourth, most of the economy of forward feed will be obtained. If it is then pumped to the first effect, there will be some sacrifice of economy due to the resulting heating in the first effect, but the final concentration will take place under conditions most favorable in the light of the properties of the solution. If, in pumping from the last to the first effects, the solution is pumped through one or two heaters fed with vapors from the first or second effects, the maximum economy will be secured.

This method of mixed feed is very often used on such cases. It involves only one more pump than forward feed, and usually gives the optimum compromise between maximum capacity and maximum economy. Moore[7] gives the results of several cases of mixed feed, including such feeds as 6, 5, 4, 2, 1, 3 in a sextuple effect. He does not give any clue to his method of solution, so that the number of arbitrary assumptions in his calculations (and hence their general applicability) cannot be determined.

Continuous Operation vs. Batch Operation.

Continuous operation is most desirable from many points of view, and is used wherever conditions are constant. The sooner the operator learns to set his valves so that they do not have to be changed except at infrequent intervals, the more generally satisfactory and economical the operation will be. The results are an even load on the boilers, evaporator auxiliaries, water pumps, etc.; a uniform discharge of material of uniform density; and the maintenance of whatever set of conditions experience shows to be the optimum for economy and capacity. The practice of some operators of drawing in a large amount of feed, shutting off the feed entirely and boiling down a foot or two, and then feeding again rapidly, is to be discouraged if only because it requires constant attention to maintain a uniform discharge.

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[7] *Chem. Met. Eng.*, *18*, 186-92 (1918).

In continuous operation, however, each effect operates on liquor at the discharge concentration for that effect, and the effect from which thick liquor is withdrawn operates on liquor of the final concentration. If this final concentration calls for a product of excessive density, excessive viscosity, or excessive elevation in boiling point, the whole apparatus is then continually working under this handicap. Under these conditions batch operation may be desirable.

If the concentration is done in single effect, batch operation is the only method permissible for material of these properties. If the concentration is carried out in multiple effect, the multiple effect may operate continuously, but discharging a product not up to the final concentration. A single effect may then take this intermediate product and bring it up the rest of the way in batches.[8] Since, as previously shown, the last few per cent increase in solid content of the solution calls for the removal of a small fraction of the total water to be evaporated, this does not cause an undue loss of economy. The point at which to pass to batch operation can only be determined by experience for each particular material, since the data now available do not permit such calculations for any practical case.

In double effect evaporators working on such materials, especially materials of large elevation in boiling point, the vapor lines may be so arranged that the bodies can be operated as a double or as singles. The evaporator is operated as a double effect, feeding either forward or backward, till one effect is full of material at such a concentration that the capacity is cut down to the limit at which double effect operation is economical. The valves in the vapor lines are then shifted and the denser material is concentrated in single effect up to the final density, feeding from the other effect which is also being operated as a single and fed with fresh liquor. Fresh liquor may be fed to both effects and both brought up to the final density in single effect operation.

OPERATING TEMPERATURE RANGE.

It would seem, at first glance, as though an evaporator should be run at the maximum total temperature drop. That is, the steam pressure to the first effect should be high, and the boiling point in the last effect as low, as possible. This is not necessarily, or even often, the case, as will be shown below.

Steam Temperatures.

The maximum steam temperature to be carried is often set by conditions outside the control of the designer or the operating engineer.

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[8] McGregor, U. S. Pat. 1,068,789, 1913.

The subject of steam economy will be discussed in Chapter 14; but it is fairly obvious that the most economical steam for operating an evaporator is exhaust from prime movers generating power. This usually determines in advance the steam pressure available.

Where direct steam is used, the designer may set the pressure for evaporators at any point he pleases, subject to the mechanical limitations of the design. Within these limitations, the highest possible steam pressure should be used; both because of increased temperature drops, and increased coefficients at higher drops and higher boiling points. The only other limiting condition is any possible damage to the liquor through exposure to too high temperatures.

On the mechanical side, a vertical evaporator may be more easily designed for high pressures than a horizontal, due to its cylindrical shape. When made in plate steel or any other reliable material, there is practically no limit to the pressures for which vertical evaporators may be designed, though the cost of the machine obviously increases with the pressure. Cast iron, because of its unreliability under tension, is not suited for high-pressure machines, and cast iron bodies are rarely built for pressures over 15-20 pounds above atmospheric. This varies with size; small evaporators can be built for higher pressures. When cast iron evaporators of the standard or central downtake type are to be used, the steam is contained in a space bounded by a cast iron ring, and the above pressure is therefore the limit of the steam pressure which may be used. When the basket type of vertical evaporator is used, the basket is usually of plate steel or copper even if the body is cast iron. In this case the above limits determine t_1 only, and t_0 may be made considerably higher than corresponds to the above pressure. In general, any type where the parts under pressure are cylindrical (Yaryan, Kestner) are subject to the above discussion.

Any type of horizontal evaporator is limited as to the maximum pressure for which it can be designed. Such types as Zaremba and Buffalo, with cylindrical bodies, are usually made of cast iron and are therefore subject to the above limitations. The Swenson type may, by internal bracing, be built for nearly as high pressures in the vapor space. The principal difficulties are found in the steam chests; which, in any type, are rectangular and therefore difficult to design for high pressures. The tube sheet is almost always cast iron, unstayed, and therefore very weak. Any evaporator, though working with a pressure difference corresponding to $(t_0 - t_1)$ across the tube sheet of the first effect, must be designed to stand a pressure difference between steam pressure and the atmosphere. This may or may not be greater than the working pressure, but if it is the greater, it is this which limits working steam pressures. Horizontal evaporators are rarely designed for over 25 pounds steam pressure.

Final Vacuum.

Many engineers are under the impression that the final vacuum should be as high as possible. A superficial examination of Figure 76 shows that at low boiling points the coefficient in the last effect decreases at least as rapidly with temperature as it increases with temperature drop. At such drops as would occur in a double or a triple, it decreases twice as fast with temperature as it increases with temperature drop. Hence, a lowering of the boiling point in the last effect is of little value; and if it be lowered very far, the evaporator actually decreases in capacity with increased total temperature drop, due to a lowering of the boiling points in all effects but especially in the last effect.

On the other hand, increasing the total temperature drop by decreasing the boiling point rapidly increases the cost of operation. The size of vacuum pump needed is increased, power for its operation is increased, and after a certain limit is passed, a more expensive type of pump may be necessary. Increasing the size of the vacuum pump often increases only its first cost; for if the evaporator is to be operated by steam at pressures suitable for the exhaust from a steam-driven vacuum pump, its exhaust simply decreases the amount of direct make-up steam used. If this is not the case, the power for the larger pump is obviously a charge against the evaporator.

The more important factor is the cost of water. Since one pound of water can absorb only a few heat units in most cases, decreasing the boiling point in the last effect by only 10° F. may decrease the heat absorbed per pound of cooling water by 15 to 25 per cent, and hence increase the weight of cooling water used by the same amount. The importance of this factor naturally varies with the cost of water, but it can seldom be neglected entirely.

All in all, it is not often that it is practical to operate the last effect of an evaporator at pressures less than three or four inches absolute (26 to 27 inches vacuum referred to a 30-inch barometer). Twenty-six inches will be found to be the vacuum held in the average plant. Here evaporator practice is diametrically opposed to power plant practice, where in the operation of modern turbo-generators, every possible device is employed to decrease the absolute pressure in the condenser.

REMOVAL OF CONDENSATE.

The mathematical treatment of the heat in condensate from an evaporator is no different in principle from the theory in Chapter 10. In practice, it is difficult to include this in evaporator problems because of the uncertainty of its temperature. This will vary with the size and design of the evaporator, as such factors vary the time the condensate is in contact with the tubes. Due to the small temperature drop in the first effect, the first effect condensate is usually but little below steam temperature. In the later effects, it is at some temperature intermediate between steam and liquor. Because of this uncertainty the problems of Chapter 10 were simplified by assuming that it left at steam temperature.

The condensate from the first effect is ideal for boiler feed, as it is hot distilled water equal in weight to the steam used. It should obviously be returned to the boilers as directly and simply as possible.

The treatment of condensate from later effects is subject to many considerations, especially considerations as to whether or not any other part of the process can use warm water of such temperatures, and also whether or not the condensate is contaminated by entrainment or leaks. If the condensate from any effect can be so used, the simplest and most economical method is to divert it to such uses as hot as possible. Condensate from the second effect, if free from contamination, may be used as boiler feed make-up.

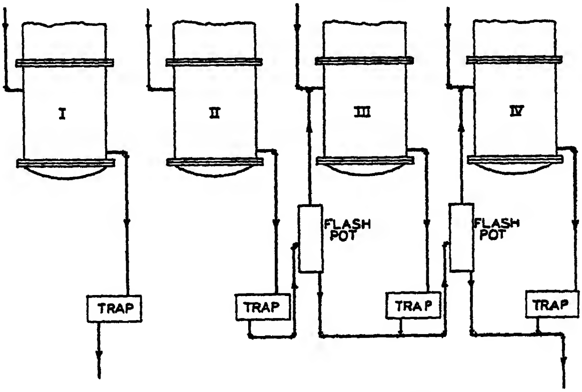

FIG. 81.—Flash pot connections.

If the condensate cannot be used for any such purposes, all the heat it contains when it leaves the evaporator is lost, and should be recovered as far as possible. One method which offers considerable advantages on paper is to pass the condensate from effect to effect, flashing it down to the temperature of the steam to the last effect.

If the condensate is actually introduced into the steam space of a succeeding effect, there is likely to be difficulty in removing non-condensed gases. The steam space of the later effects will be flooded with undue amounts of condensate. The best method is to place receivers of sufficient capacity, called flash pots, under each effect and connected into the steam space of that effect. Condensate from all preceding effects except the first is trapped or pumped into these flash pots in succession. Figure 81 is a diagrammatic sketch of such an arrangement.

It should be noted that condensate from the second effect evaporates (n — 2) times the weight of its flash; from the third, (n — 3) times, etc. Condensate from the last effect cannot be so flashed, and hence the method is of limited application. Considering further that owing to its temperature being less than that of the steam from which it it is formed, condensate from any effect flashes off considerably less vapor than corresponds to the temperature drop between effects, the advantages are seen to be very small except where a large number of effects are used on a large total temperature drop. The saving seldom warrants the piping complications.

It should be noted that first effect condensate should never be flashed in this manner. So far as the evaporator itself is concerned, since this flash is used for (n — 1) effect evaporation, an appreciable saving in steam to the evaporator may result. But if this condensate is to be used as boiler feed, it must be heated from t_{n-1} to t_0 in the boiler. This heat used to heat the condensate back to t_0, if used in the evaporator would produce n-effect evaporation, which is a more economical use of the heat than (n — 1) effect evaporation from flash. In other words, the steam economy of the whole plant is adversely affected by this process which, calculated for the evaporator alone, affects the evaporator favorably.

Warm condensed water which is not to be used for boiler feed may be very simply used in heat exchangers for heating cold incoming feed or for any other purpose, if the amount of heat it contains justifies the expenditure.

REMOVAL OF SOLIDS.

In the concentration of many materials, solids are separated in the evaporator, either as primary or incidental products, and must be removed during the operation. Methods for accomplishing this may be classified as follows:

Dumping the charge
Elevators
Salt receivers
Settlers

Dumping the Charge.

This is used on extremely small scale operations of all kinds, but is regularly used on the large scale only in the sugar industry. Here the crystals are small and the mother liquor is so extremely viscous that no settling of the crystals can occur during the operation. In the case of any other material of similar properties it would be the most practical method.

Elevators.

Elevators are confined to the salt industry. The evaporator below the steam space is cone-shaped, to provide a space in which crystals may settle. Salt evaporators are usually fed in parallel with saturated brine, and hence seed crystals form at once in all effects. They are held in suspension by the agitation due to boiling until they become large enough to settle in spite of this agitation. The cone is prolonged into a vertical or slightly inclined pipe which extends such a distance below the normal liquor level as to form a barometric seal. This pipe is usually called the "salt leg." At the end of this pipe is placed the boot of a bucket elevator. This elevator must discharge at a point above the normal liquor level, and it must be enclosed up to the liquor level, so that the vacuum may be broken without draining the evaporator. The buckets are usually of wire screen or perforated metal, so that the salt may drain as it is elevated. Frequently the elevator chains run in grooves in the casing so that rust or dirt from them may not fall into the salt so easily. The product coming from these elevators will usually contain 50 per cent moisture, and must be stored in drainage bins or sent through a centrifugal.

The feed to the evaporator may be introduced at the elevator boot. If so, the rising current of brine through the salt leg acts as a sizing device to wash out fine grain. The product of a salt evaporator is usually so uniform that this device is often dispensed with, and the feed introduced at any convenient point.

Elevators are rather expensive and consume considerable power. In the salt industry, where the evaporator usually takes many times as much steam as is produced in the form of exhaust from all power units, this is not a serious item. Elevators are rather dirty, especially on material like salt which corrodes rapidly during shutdowns. They are continuous and require very little attention.

Salt Receivers.

Probably the method of most general application is to connect to the bottom of the evaporator one or more receivers which may or may not be equipped with filters. These vessels may be cut off from the evaporator by valves, and have a connection to the vapor space of the evaporator so that they may be put under the same pressure as the evaporator before being reconnected.

If provided with a filter bottom they have a connection below the filter for the pump which removes excess liquor, a connection on top for air, steam, or water for washing or drying the precipitate, and a door or other means for removing the salt. A typical construction is shown in Figure 82. If they do not have a filter, the whole charge may be dropped into an open filter box, or pumped to a filter of any desired type.

Settlers.

Where large amounts of solids are to be removed, and where the dirt and waste of elevators are undesirable, some form of settler is used. In this method the bottom of the evaporator is connected to a pump which removes continuously a mixture of liquor and precipitate. This is pumped to a settler in which a cross-section is enlarged so that the precipitate may settle out, while the liquor with small crystals in suspension is returned to the evaporator. Each effect may have its own settler, the evaporator body and the settler operating as a closed system, or there may be one open settler for all effects.

Such settlers are usually cylindrical vessels with conical bottoms. The diameter of the cylinder is such that the volume of liquid being removed by the salt pump when passing up through such a cylinder will have a velocity low enough to allow crystals of the desired size to settle. If the crystals are to be withdrawn into centrifugal machines throughout the operation, the settler is most simply an open settler. If the apparatus operates on the batch system and crystals are not withdrawn till the end of the batch, the settler may operate on a closed system, i.e., it may be connected directly to the evaporator for the return of liquor, and operate under the pressure existing in the evaporator body.

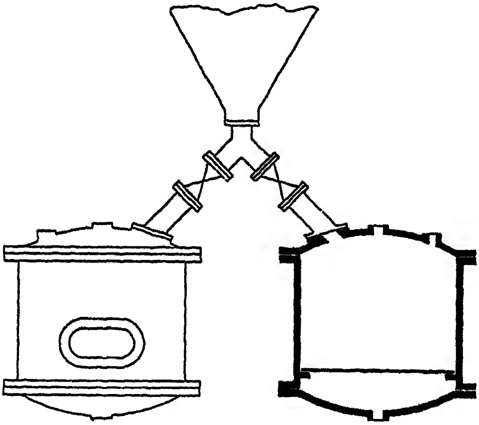

FIG. 82—Salt fillers

The general principle of continuous removal of crystals was patented by Trump.[9] There are a few patents on systems of salt receivers, but none contain any device of special value.[10] In the case of a closed settler operating on the batch system, the crystals may pack in the cone and be difficult to withdraw. Wheeler[11] has patented the idea of introducing the stream of salt and liquor at the point of the settler cone, instead of somewhere in the cylindrical part. This keeps the salt stirred up and makes its final discharge simple. Pritchard and Gallop[12] have described the operation of such settlers on a caustic evaporator.

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[9] U. S Pat. 743,351, 1903.

[10] Ordway, U. S. Pats. 775,577, 1904; 905,568, 1908; Kestner, U. S. Pat. 1,191,108, 1916.

This method of removing crystals is coming into favor quite rapidly. It is a simple, self-contained, and clean method. It uses the minimum of labor. The principal problem is the maintenance of the pumps, for this service is obviously difficult. The mixture pumped varies from one part salt to two parts liquid by volume, up to one to ten. One to one mixtures can be pumped but it is not safe to design on this basis. One to three or four is probably a safe operating figure. For such work the pumps must be exceedingly rugged, with over-size shafts, extra long stuffing boxes, and provision for feeding water or clear solution into the packing. Crystals carried into the packing by leakage will cut through a bronze shaft of the average diameter in a couple of days. It should be possible to renew parts where wear may occur with the minimum of trouble. Pipe lines should be as straight as possible, with no pockets where crystals might collect. Pipe line velocities should be kept high—five feet per second or over. Rubber dredge hose has been used for this service, and makes the finding and removing of plugs very simple.

Numerous attempts have been made to devise a continuous salt removal valve. These often take the form of a horizontal rotating disc, provided with a number of salt pockets. As the disc rotates these pockets come under the evaporator cone and fill with salt, then at a point about 180° in their rotation they discharge into a receiver. Such devices are rarely satisfactory. They must operate between guide plates which are supposed to be vacuum-tight on the side where they connect with the evaporator, but salt crystals get in between the disc and these guides and wear them so that leaks are the rule. Further, every pocket coming under the evaporator cone is full of air as it comes in, and thus considerable amounts of air are admitted to the evaporator even if the valve guides are tight.

External Crystallization.

If a substance is to be separated from solution by boiling off water, the first thought is usually to use a salting evaporator and remove the crystals by some one of the above schemes. If the substance is very soluble and has a steep solubility curve (i.e., if its solubility increases rapidly with temperature), such a procedure may lead to unpleasant results.

Consider an evaporator under vacuum crystallizing such a substance—say, sodium phosphate or ammonium chloride. The solution from which crystals are separating is very dense and possibly viscous. A little irregularity in operation slows down the circulation, or steam pressure falls off for a few minutes. The rate of heat transfer is decreased, the weight of steam evolved is decreased; but if water to the condenser is not adjusted or the vacuum pump slowed down, the absolute pressure may be lowered. An increase in vacuum from 26 inches to 26.5 inches corresponds to a lowering of boiling point of 5° F. The liquid flashes, becomes colder and more viscous, the solubility decreases, a number of fine crystals are thrown out, and thus circulation is still further decreased and the rate of steam evolution cut down. The effect is cumulative and rapid, and in a minute or two the evaporator may freeze solid This is especially true if the salt crystallizes with much water of crystallization.

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[11] U. S. Pat. 1,222,340, 1917.

[12] *Ind. Eng. Chem., 16,* 1058-59 (1924).

Such materials are best handled by concentrating in an evaporator, removing concentrated solution continuously, crystallizing in an external crystallizer, and returning the mother liquor to the process.

SCALE.

Formation.

In order that true coherent scale may form on a heating surface, it is necessary that there be present some substance whose solubility decreases with increase in temperature. For most substances the solubility increases with increasing temperature.

If a solution of a substance whose solubility curve is normal is concentrated, any material thrown out of solution must obviously come from the cooler part of the solution where its solubility is the least. This means that crystals must form and grow in the bulk of the solution, and that these crystals can only adhere to the heating surface under unusual conditions. If a saturated solution of such material is heated, it cannot deposit any solids.

On the other hand, if a substance be present whose solubility curve is inverted, the hotter film next the heating surface is the part where the solubility is least, where saturation will first appear, and where solids will first deposit. Most solubility curves which decrease in temperature are very flat; i.e., they call for a relatively small change in solubility for a given change in temperature. The effect of a stagnant film and a low degree of super-saturation tend to the formation of a dense and uniform layer of crystals. Any pits or irregularities in the heating surface contain the most highly superheated solution and hence are the areas where crystallization must begin. This results in bonding the scale to the surface. If at any one point growth should be more rapid than the average, the crystal so formed would project through the stagnant film into colder layers, where the material in question is more soluble, and hence growth would stop. The result is a hard, coherent, smooth film of scale. This will obviously occur whether the solution is evaporated or heated, provided that the solution is nearly enough saturated to reach saturation at film temperature.[13]

If, during the process of scale formation, solid matter is present in suspension, it will be included in the scale and increase the rate at which it forms. Material in solution will often be occluded also, so that scales formed in practice are quite complex in composition. In the absence of a substance having an inverted solubility curve, true scale cannot form.

The commonest substance of this class is, of course, calcium sulfate. It occurs almost universally in natural waters and brines; and in many commercial solutions, from the use of sulfuric acid or sulfur dioxide and lime in the process. Consequently, most scales found in commercial operations are mainly calcium sulfate. The solubility relations of calcium sulfate[14] are shown in Figure 83. Gypsum ($CaSO_4, 2H_2O$) is the common form at ordinary temperatures. In aqueous solutions its solubility curve shows a maximum at about 90° F. This point will be shifted somewhat by the presence of other materials in solution, but in general it lies too low for commercial evaporators to operate on the rising branch of the curve.

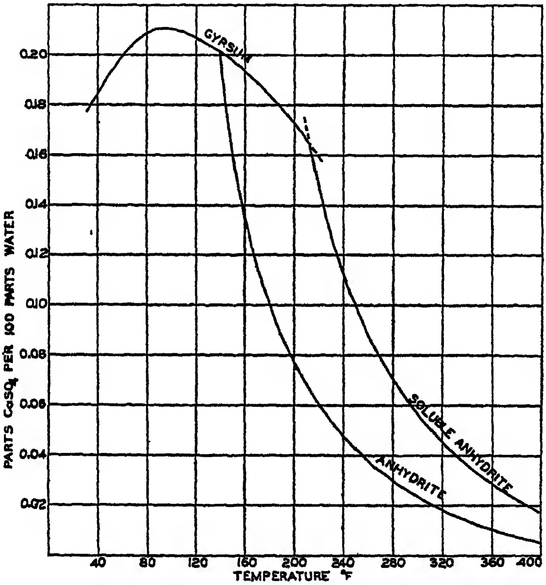

Fig. 83.—Solubility of calcium sulfate.

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[13] See Hall, *Ind. Eng. Chem.*, *17*, 283-90 (1925).

[14] Melcher, *J. Am. Chem. Soc.*, *32*, 62-5 (1910).

Consequently calcium sulfate will always be a scale-forming material.

At temperatures above 160° F. anhydrite ($CaSO_4$) is the stable form. This occurs in two modifications, one more soluble than the other. The stable form is the less soluble one, and hence this is the constituent most commonly found Although the solubility curves of anhydrite are steep, the absolute differences in solubility are small, and hence the conditions for uniform growth of scale are still maintained.

Calcium carbonate is sometimes apparently a major constituent of scale; but in the complete absence of calcium sulfate, no scale, or only a very soft one, is formed Under the minimum possible vapor pressure of CO_2 (conditions occurring in a boiler or evaporator) the solubility of calcium carbonate decreases with temperature and hence it cannot be a scale-former. If it exists in a water containing much carbon dioxide in solution, as the water is heated the CO_2 content decreases and hence the solubility of calcium carbonate decreases. Under these conditions it may form scale, and these conditions occur in feed-water heaters and boiler feed lines.[15] The scale is always soft, however; and in many cases the total absence of calcium sulfate prevents any scale formation, the calcium carbonate depositing as a fine mud.

Several substances have such inverted solubility curves and are therefore scale formers. Their number is small compared to those having normal solubility curves. Anhydrous sodium sulfate and sodium carbonate monohydrate are examples.[16]

Rate of Scale Formation.

McCabe and Robinson[17] have developed the following theory: The heat transferred in an evaporator is given by

$$Q = UA\Theta T \tag{78}$$

where U is the overall coefficient of heat transfer, Θ is the temperature difference, and T is the time. From this,

$$\frac{dQ}{dT} = UA\Theta \tag{79}$$

If we consider that U is a compound conductance, made up of the conductances of the steam film, metal, water film, and scale; and if we assume that the two fluid films do not change as scale forms, we may write

$$U = \frac{1}{\alpha + BQ} \tag{80}$$

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[15] Hall, *loc. cit.*

[16] Badger and Caldwell, *Trans. Am. Inst. Chem. Eng.*, *17* (I), —— (1925).

[17] *Ind Eng Chem*, *16*, 472-3 (1924).

where α is the combined resistance of the water film, metal, and steam film, and B is a constant. This is equivalent to stating that the thickness of the scale (which is proportional to its resistance) is directly proportional to the total amount of liquid which has been evaporated. From this

$$Q = \frac{1}{UB} - \frac{\alpha}{B} \tag{81}$$

and

$$dQ = -\frac{dU}{BU^2} \tag{82}$$

Substituting in (79)

$$-\frac{dU}{BU^2} = UA\Theta dT \tag{83}$$

and

$$-\frac{dU}{U^3} = BA\Theta dT \tag{84}$$

from which

$$\frac{1}{U^2} = 2BA\Theta T + C \tag{85}$$

Since $2BA\Theta$ is a constant, equation (85) reduces to

$$\frac{1}{U^2} = aT + b \tag{86}$$

This has been checked by McCabe and Robinson, and by Badger and Othmer[18] against a variety of data and found to express actual operation very well. It is only necessary, therefore, in order to determine the capacity, after any desired time, of an evaporator in which scale is forming, to make two determinations of the rate of heat transfer at known times, plot $\frac{1}{U^2}$ against T, draw a straight line, and read off $\frac{1}{U^2}$ at any desired time.

SCALE REMOVAL AND PREVENTION.

Methods for removal or prevention of scale may be classified as follows:

Mechanical:
- Cutters or cleaners
- Scouring methods
- Deformation of the heating surface

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[18] *Trans. Am. Inst. Chem. Eng.*, *17* (I), —— (1925).

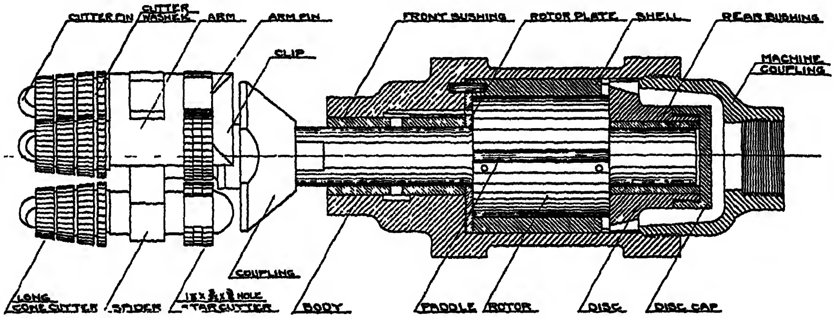
Chemical:
- Purification of the liquor before evaporation
- Dissolving the scale
- Furnishing seed crystals

Electrical

Tube Cleaners.

Tube cleaners are the most familiar devices in this connection, as they have been well developed in steam boiler practice. They may be designed to cut the scale from the inside of a tube; or to pass inside a tube which has scale on the outside, removing the scale by cracking it off. This latter is really a case of mechanical deformation of the heating surface.

Vertical tube evaporators, or any other type in which the liquid is on the inside of the tube, are easily cleaned by this method. Such tubes are usually at least 1¼ inches in diameter and more often over 2 inches. Cleaners of this type are usually driven by a small rotary motor operated by steam or air, which may be made small enough to pass inside the tube with the cutter head (Figure 84). Such turbines cannot be made for tubes less than 1¼ inches, and are most satisfactory for tubes over 1½ inches. For smaller tubes the cutter must be on a rigid shaft driven by an external motor.



Courtesy Lagonda Mfg. Co.

FIG. 84—Turbine tube cleaner.

The turbine driven cleaners run at very high speeds, do not have much power, and operate essentially by cracking the scale. In many operations where the scale is heavy it is more rapidly removed by using a cutter at slower speeds with more power, and actually drilling out the scale rather than cracking it off. Such drills may be driven by air or steam motors, which are lighter than electric motors. Where the workman must get inside the evaporator with the motor, this limits the tubes to 4 feet 6 inches in length, as he cannot lift the motor and drill much higher. In the case of heaters which have to be drilled at frequent intervals, it may be desirable to rig a permanent guide and hoist to handle the drill and motor. Drill points are made like Figure 85.

Either type of cleaner is satisfactory for iron or steel tubes, but both are rather hard on copper or brass tubes. A cutter head may be obtained with a limit ring, which limits the travel of the cutter arms so that the metal is not cut, but this cannot be made for tubes less than 2½ inches inside diameter.

For evaporators in which the liquid is outside the tubes, the rattler type of cleaner is seldom available, as horizontal tubes are too small to take the turbine, and rattler heads driven from a fixed shaft cannot be made any smaller than for turbine drive. There is no satisfactory method for removing hard scale from the outside of a small tube except to take the tubes out and chip them by hand or sandblast them. This fact alone indicates the limitations of the horizontal tube evaporator. Sometimes soft mud can be removed from the outside of horizontal tubes by motor driven brushes, but this is rarely satisfactory. Ease of cleaning is an important advantage of the Yaryan type, as here the scale is inside the tubes, but the tubes are large enough to take the turbine type cleaner.

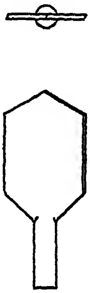


FIG. 85.—Drill point for heavy scale.

Scouring.

Scouring is rarely practiced, but in some forced circulation evaporators like Figure 71, clean silica sand or other insoluble material is introduced into the liquid to wear off scale as fast as it may be formed.[19]

Deformation.

Deformation is regularly practiced in certain types of evaporators. In the Lillie, a multiple effect may be built with vapor and liquor pipes in duplicate, two condensers, and such valves that the evaporator may be reversed, i.e., the first effect made the last and the last effect the first[20] This changes the temperature in all the effects; and since the Lillie tubes are supported at one end only, they are free to expand or contract with changes in temperature. If the evaporator is working on a material which deposits calcium sulfate or other hard, brittle scale, this change in the tube cracks off the scale fairly well It does not affect a soft scale. If the solution is anywhere nearly saturated with calcium sulfate the scale will form mainly in the first effect, and hence on reversal these tubes will suffer the greatest change in temperature and will exert the maximum cracking effect.

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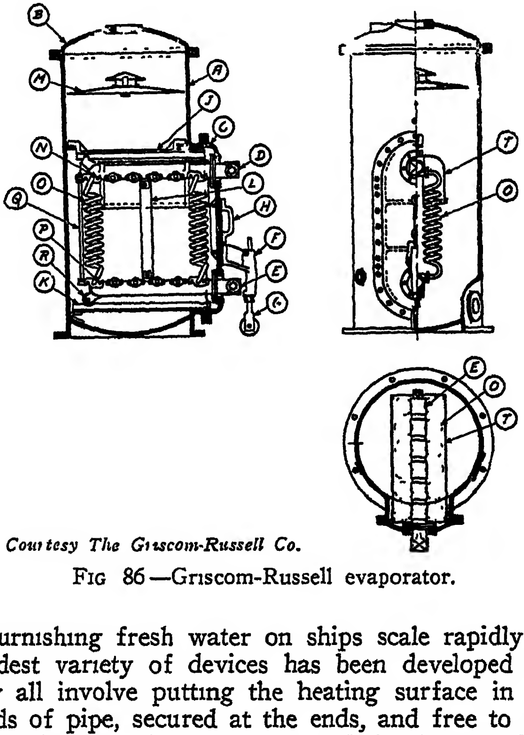
[19] Prache and Bouillon, Engl. Pat. 137,052, 1919.

Since sea water is practically saturated with calcium sulfate, evaporators for furnishing fresh water on ships scale rapidly, and in this field the widest variety of devices has been developed for cracking scale. They all involve putting the heating surface in the form of coils or bends of pipe, secured at the ends, and free to deform more or less under changes of temperature. This change of temperature is usually brought about by cutting off the steam and filling the coils with cold water. Some of these evaporators were developed especially for cracking off scale; others were developed without this idea but are suitable for carrying out the process.[21]



Courtesy The Griscom-Russell Co.

FIG 86—Griscom-Russell evaporator.

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[20] Leilich, *Met. Chem. Eng.*, *15*, 213-4 (1916); Lillie, U. S. Pats. 777,114, 1904; 948,376, 1910; 984,226, 1911.

[21] *Flat coils*—Weir, U. S. Pat. 1,049,014, 1912; Braun, U S. Pat. 1,334,014, 1920; Fothergill, U S Pat 1,498,440, 1924; Morrison, Engl. Pats. 13,775, 1900;

One of the best known of these coil evaporators is the Griscom-Russell (Figure 86). Attached to the shell A is a door C, carrying cast iron manifolds D and E. To these manifolds are connected coils of copper tubing O. The door carries one or more brackets F with rollers G, and the manifolds run on rollers R between guides J and K. Over the coils is a baffle T. The entire heating unit may be withdrawn for cleaning.

When the coils are coated with scale, steam is cut off and cold water turned into them. The resulting contraction clears the coils of scale if the scale be hard and brittle, and not too thick. None of the devices depending on distortion of the heating surfaces are of value on soft scales.

Purification.

Purification of the liquid before sending it to the evaporators is an obvious method of removing scale but one that is not often economical. Calcium sulfate may be completely removed by treatment with sodium carbonate. Hall[22] has shown that in the presence of sufficient carbonate ions, precipitation of calcium sulfate is impossible; and in the papers cited he shows the limiting concentrations of carbonate ions which must be present at various temperatures, to prevent calcium sulfate scale.

This method, while often used for treating boiler waters, fails in many cases occurring in practice because of the expense involved. To precipitate the calcium sulfate from 1000 gallons of a liquid containing 0.1 per cent of $CaSO_4$ would take 6.5 pounds of sodium carbonate without using any excess. The cost of reagent alone is often sufficient to make purification impossible, without considering the cost of tanks, attendance, and sludge disposal. The possibility of scale prevention by chemical means, as compared to scale removal, should always be examined carefully before a process is adopted.

Dissolving Scale.

Dissolving scale which has already formed is a practice which is almost the only one possible when using a horizontal tube evaporator on a material which scales. In the sugar industry (both beet and cane) there is present in the juice not only calcium sulfate, but

21,282, 1901; 10,812, 1905; 24,074, 1909; 23,403, 1912; Weir, Engl. Pat. 2,812, 1912; Fr. Pat. 444,948, 1912. *Spiral coils*—Zastrow, U. S. Pats. 964,358, 1910; 1,124,096, 1915; Power, U. S. Pat. 1,028,777, 1912; Row, U. S. Pat. 1,131,738, 1915; Evans, U. S. Pat. 1,225,118, 1917; Benjamin, U. S. Pats. 1,225,502, 1918; 1,280,641, 1918; 1,288,480, 1918; 1,302,625, 1919; 1,355,935, 1920; Jones, U. S. Pat. 1,299,955, 1919; Row and Davis, U. S. Pat. 1,304,379, 1919; Dyson, *et al.*, U. S. Pat. 1,440,723, 1923; Dyson and Stuart, U. S. Pat. 1,477,328, 1923; Brown, U. S. Pat. 1,501,646, 1924; Griscom-Russell Co., Engl. Pats. 103,820, 1916; 217,563, 1924; 218,248, 1924. *Special devices*—Field, U. S. Pats. 1,451,901, -2, -3, -4, 1923; 1,480,382, 1924; Sauerbrey, Fr. Pat. 444,441, 1912.

[22] *Ind. Eng. Chem.*, *17*, 283-90, 409-11 (1925).

calcium salts of other acids which also have inverted solubility curves, such as calcium acetate, citrate, and malate. Even when vertical tube evaporators are used, in many sugar mills it is customary to boil out rather than drill off the scale.

The usual reagents used are sodium carbonate and sodium hydroxide, alone or mixed, followed by hydrochloric acid.[23] Calcium sulfate is very slightly soluble, but it is more soluble than either calcium carbonate or calcium hydroxide. Prolonged boiling with sodium carbonate or sodium hydroxide converts the compact sulfate scale to a soft carbonate or hydroxide scale, which may be scraped off or dissolved with dilute hydrochloric acid. Evaporators built of iron or steel may be boiled out with 0 5 per cent HCl with little danger of attack on the metal if the boiling is not prolonged after the scale is all dissolved.

Sodium bisulfite[24] has also been used, and for scales containing much silica, ammonium fluoride.[25] Acetic acid formed by fermentation has been suggested.[26] In some cases it has been suggested that the liquid to be heated or evaporated be made acid,[27] but this cannot prevent the formation of calcium sulfate scale, and slight overdosage of acid might seriously damage the apparatus.

Seed Crystals.

If the film of supersaturated liquid which is the cause of scale be bombarded with a large number of seed crystals of the scale-forming material, and if the seed crystals move with such velocity that they do not remain in the stagnant film to be entrapped in the scale, scale formation may be greatly diminished, if not completely prevented. So, if calcium sulfate crystals in large numbers are added to a solution which is already saturated with calcium sulfate and vigorously circulated, material which would otherwise form scale deposits on these crystals instead.[28]

A similar idea has been used for the evaporation of such substances as anhydrous sodium sulfate, and sodium carbonate monohydrate which have inverted solubility curves.[29] In these cases the method used is to withdraw continuously a part of the solution in the evaporator, superheat it in an external heater, and return the superheated liquid under the heating surface. The resulting flash evaporation provides both vigorous stirring, which keeps down the thickness of the stagnant film, and a large number of seed crystals. This does not entirely prevent scale, but delays its formation enough to make possible a cycle of such length as to be of commercial value.

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[23] Wendeler, *Deut. Zuckerind*, *32*, 420-22 (1907); Peck, Bull 33, Hawaiian Sugar Planter's Ass'n (1910); Pellet, *Deut. Zuckerind.*, *35*, 596-8 (1910); Peck, *Int. Sug. J.*, *13*, 357 (1911); Thurlow, *Int Sug. J.*, *14*, 328-31 (1912); Fallada, *Oest.-Ung. Z. Zuckerind*, *41*, 512-6 (1913); Bolk, *Arch. Suikerind.*, *24*, 1194-7 (1916), Anon., *Int. Sug. J.*, *24*, 142 (1922).

[24] Kettler, *Deut. Zuckerind*, *32*, 219-21 (1907), Schimmel, *Arch Suikerind*, *24*, 1607-8 (1916).

[25] Anon., *Arch. Suikerind*, *31*, Verslag 4, 171-7 (1923); Vander Linden, *Int Sug. J.*, *25*, 637-40 (1923).

[26] Hartmann, *Oest -Ung. Z. Zuckerind*, *40*, 715-7 (1911).

[27] Holle, U. S. Pat 1,405,783, 1922; Martin, U. S Pat. 1,447,096, 1923

[28] Bull, U S. Pat. 1,399,845, 1921, Aktieselskabet de Norske Saltwerke, Engl. Pat. 131,279, 1919; Norw Pat. 30,347, 1920.

[29] Badger and Caldwell, *loc. cit.*

Electrical Methods.

Recently considerable publicity has been given to a method of scale prevention in which a thermocouple is connected to the surface on which scale may deposit, and to an insulated electrode in the solution, so that the heating surface is the cathode.[30] A current of a few milli-amperes is said to be sufficient to prevent scale, or gradually decompose scale already formed. The only possible method by which such devices can operate is a slow evolution of hydrogen at the cathode. No authoritative data have been published regarding them.

FOAM AND ENTRAINMENT.

These two terms are often loosely used—sometimes almost interchangeably. It is necessary in the following discussion to use them in quite definite senses. *Foaming* is the formation of a more or less permanent, continuous, layer of bubbles, which covers the whole surface of the liquid, and is due to some specific property of the liquid being evaporated. *Entrainment* is the mechanical carrying off of drops of liquid due to the velocity of the current of vapor into which they may be projected. Any liquid may suffer entrainment depending on the violence of boiling and the velocity and direction of the vapors.

Foam.

The formation of foam has seldom been discussed systemically, and many false opinions exist as to its cause. The best general statement of the cause of foam is by Bancroft[31] as follows:

"To get a foam the only essential is that there shall be a distinct surface film, in other words, that the concentration in the surface layer shall differ perceptibly from that in the mass of the liquid. All true solutions will therefore foam if there is a marked change of surface tension with concentration, regardless whether the surface tension increases or decreases. All colloidal solutions will foam if the colloid concentrates in the interface or if it is driven away from the interface. To get a fairly permanent foam the surface film must either be sufficiently viscous in itself or must be stabilized in some way. This can be done by introducing a solid powder into the interface.

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[30] Brunker, *Deut. Zuckerind.*, *46*, 759 (1921); Renger and Fuhrmann, Engl. Pats. 154,610, 1920; 173,418, 1920; Schnetzer, Engl. Pat. 174,905, 1921; A.-G. für Chemische Industrie in Liechtenstein, Fr. Pat. 556,860, 1923.

[31] "Applied Colloid Chemistry," pp. 269-70, McGraw-Hill, 1921.

"Solutions of aqueous alcohol, acetic acid, sodium chloride, and sulfuric acid all foam when shaken, but the foam is instable Soap solutions foam when shaken and the foam is, or may be, quite stable owing to the viscosity of the soap film. With saponin the surface film is even more stable. If we add to aqueous alcohol some substance like lycopodium powder which goes into the interface, we get a stabilized foam. We can do the same thing with aqueous acetic acid by adding lamp-black. The presence of enough of a finely divided solid in the interface will make a film so viscous that the foam will be quite stable. Grease will help stabilize a foam in some cases and it has been claimed erroneously that the foaming of sulfuric acid solutions is due to grease."

In order to explain the first two sentences of the above, the following sentences from an earlier chapter[32] are of value. "When a substance dissolves in a liquid, the surface tension between the liquid and vapor changes, increasing in some cases and decreasing in others. . . . Most salts increase the surface tension of the solution, the increase being very nearly proportional to the concentration . . . The change in surface tension when a solution is formed is also accompanied by a change in concentration, the surface film of the liquid having a different concentration from the mass of the liquid solution. The simple rule in regard to this is that the concentration in the film tends to change so as to decrease the surface tension. Consequently the surface film will be more dilute than the mass of solution if the dissolved substance increases the surface tension, and will be more concentrated than it, if the solute lowers the surface tension of the solution."

Foulk[33] has applied this theory to the problem of foaming in steam boilers, and has confirmed it in its essentials by laboratory experiments. When boiled, most solutions of inorganic salts did not foam, but the addition of powdered insoluble material, in some cases coarser than 40 mesh, caused foam by stabilizing the surface film. Insoluble finely divided material in pure water did not cause foam. Solutions of colloids which either change the surface tension of the solution or concentrate in the film cause foam. A clean boiler fed with softened water will not foam, even with fairly high concentrations of soluble salts; but a dirty boiler fed with the same water will foam because of disintegration of the scale. Clean soap lyes (glycerine-sodium chloride solutions) do not foam, but a trace of soap or fatty acid concentrates in the surface layer and causes foam. Foaming is never noticed with saturated sodium chloride solutions, or even with strong caustic solutions, in the absence of any solids or colloids to stabilize the film. In ore flotation, the oil film is not stable enough to produce foam, but the particles of sulfide ore which are wet by the oil are carried into the film and stabilize it, producing a very solid foam.[34]

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[32] "Applied Colloid Chemistry," pp 129-30.

[33] *Ind Eng. Chem*, *16*, 1121-5 (1924).

Prevention of Foam.

The subject is too complicated, and our understanding of it is still too incomplete, to enable us to derive definite rules for preventing foam. One obvious method is to remove all colloidal and insoluble material which might stabilize the film. This can sometimes be done, but often the colloid which causes the foam may be the valuable or principal constituent. If the foam is caused by a decrease in surface tension, the addition of some material which increases it, and vice versa, may prevent the formation of a film. Such methods must be worked out for each particular case, and are not subject to generalizations.

Usually the problem is not to prevent foam, but to operate an evaporator on a substance which does foam. An ordinary evaporator, either vertical or horizontal, working on a foamy liquid, may be completely emptied in a minute or two by an unskillful operator. There are three general methods of handling foamy liquids—breaking the foam by bringing it in contact with hot surfaces, by excessive velocities, or by the addition of some reagent.

If an ordinary evaporator, working on a foamy liquid, be operated with a very low liquor level, a cautious operator can prevent the whole mass boiling over, but constant attention is required. The foam must be kept low enough so that it is completely broken; and if, by too vigorous boiling, a coherent film of liquid rises on the heating surface a little above the proper level, the mass will boil over. This method usually involves not only careful control of level, but also of rate of boiling (i.e., temperature drop) and hence is especially difficult to apply to a multiple effect. If the material being evaporated deposits solids on the part of the heating surface in contact with foam only, very serious difficulties may result, especially in the horizontal tube type, where removal of such deposits is difficult.

If masses of foam are once carried up into the vapor lines, there is little chance of stopping or removing them. Foam catchers, as shown in Figure 87, if of generous dimensions, sometimes help if the amount and permanence of the foam is not too great. If there is sufficient drop in velocity of the vapors at the point where they enter the foam-catcher to cause a drop in pressure, the foam bubbles may be burst, but this is only accomplished by such a pressure drop as would appreciably decrease the working temperature drop, and is not a factor in any ordinary design.

If the liquid which tends to foam can be driven against baffles at a high velocity, the foam may be broken. In this respect the Yaryan has been especially successful on paper liquors from resinous woods where foam is serious. Probably other high-speed evaporators would also be successful in this respect, but data are not available to make this certain.

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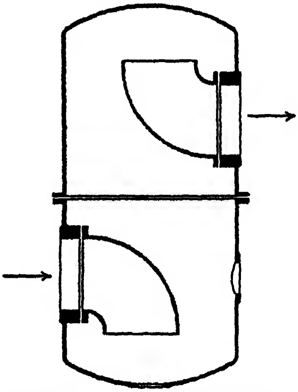
[34] Bancroft, *loc. cit.*, pp. 272-3.

The third method, of adding some reagent to break the foam, is a time-honored one in the sugar industry. Insoluble or colloidal material passing to the evaporators easily causes foam, especially in the first effect. Sugar evaporators are regularly provided with "buttercups," vessels in which oil or fat may be placed, to be drawn into the bodies under vacuum by opening a cock, or forced into pressure bodies by a plunger. Lubricating oil, lard oil, various liquid fats, vaseline, and tallow may be used.[35] Foulk[36] states that a trace of castor oil instantly stopped foaming in every experiment. If one drop of castor oil were shaken with 100 cc. of water, a few drops of the water would then instantly stop foam. Sulfonated castor oil is used in some plants for the same purpose. The cause is obscure, and cannot be due to an effect on surface tension, because it is equally effective whether or not the foam is caused by a lower or a higher surface tension than of water. Foulk suggests that it may be due to the castor oil throwing out the solid stabilizing material from the interface.



Courtesy Swenson Evaporator Co

FIG. 87.—Foam catcher

Entrainment.

The cause of entrainment is primarily the lifting of drops of liquid out of the main body by the exit velocity of the steam. Hausbrand[37] makes an elaborate mathematical analysis of this and related phenomena, but he bases his discussion on the incorrect assumption that drops of liquid leave the surface at the same linear velocity as the vapors. This cannot be true in all cases; if it were, such factors as viscosity and surface tension would have no effect, whereas in practice they do have considerable effect.

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[35] Gonnerman, *Cent Zuckerind.*, *18*, 1343-6 (1910).

[36] *Loc. cit.*, p. 1124.

[37] "Verdampfen, Kondensieren und Kühlen," 6th Ed. (1918), pp 226-60; English translation of 2nd Ed. (1903), pp 117-60.

All we can say at present is that the velocity of drops projected from the surface is some function of the velocity of the vapors. Walker, Lewis and McAdams[38] suggest that the velocity of the drops is a function of the mass velocity of the vapors (lbs. vapor per sq. ft. of path per sec.) rather than their linear velocity. This is most reasonable, for the amount of energy in the vapor available for producing velocity in the liquid is obviously a function of the mass of the vapor as well as its linear velocity.

If particles of liquid were shot into a still vapor space at a known velocity, the height to which they would rise, and hence the height of the vapor space necessary to completely prevent entrainment, could be calculated from the ordinary formulas of accelerated motion. In the first place, we do not know these velocities, even approximately; in the second place, the vapor space is not still, but contains a rising current of vapors.

If finely divided material falls freely through a fluid, the frictional resistance exerted upon it increases with its velocity. When this resistance equals the acceleration of gravity, the particle falls from then on with a constant velocity. Stokes' law states that this terminal velocity is given by

$$V = \frac{2r^2 (S - S')g}{9\eta} \tag{87}$$

in which r is the radius of a particle, S its specific gravity, S′ the specific gravity of the fluid, g the acceleration of gravity, and η the absolute viscosity of the fluid. If the fluid have an upward motion at a velocity V, the particle remains stationary; if the fluid have a velocity greater than V, the particles will be carried over with the fluid. Hence, if particles are shot up from a liquid surface into a rising current of vapor, the height to which they rise is greater as their terminal velocity approaches that of the vapor; and those which are small enough will have a terminal velocity so small as to be carried over with the vapor.

From Stokes' law we see that the amount of entrainment should increase as (1) the diameter of the drops decreases, (2) as the difference in density between liquid and vapor decreases, (3) as the viscosity of the vapors increases. Since none of these are under control, and since the size of the drops is unknown, it follows that no calculations may be made as to the height of vapor space needed to prevent entrainment. It also follows that for a certain velocity of vapor and a drop smaller than a certain limit, no extension of the vapor space, no matter how high, can prevent entrainment. For drops larger than this minimum, extensions of the vapor space will diminish entrainment.

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[38] "Principles of Chemical Engineering," p. 396, McGraw-Hill, 1923.

Obviously the initial velocities of drops ejected from the surface will be greater in a standard vertical than in a horizontal, and in a vertical they will be greater the longer and narrower the tubes. The deflector in a basket type vertical is a very efficient entrainment baffle, fulfilling the requirements discussed below. This is borne out in practice, where it is found that entrainment losses are apt to be less in the horizontal type than in the vertical type A certain amount of vapor space must be provided in any evaporator to prevent too much entrainment and allow the heavier drops to fall back. This may be anywhere from 3 to 15 feet, according to the size of the evaporator and the nature of the liquid. Experimental work connecting loss by entrainment with mass velocity of vapor, height of vapor space, viscosity, and surface tension of the liquid, is much needed.

Entrainment Separators.

Unlike foam, liquid carried along as entrained drops may be mechanically separated from vapors. If a vapor with suspended drops of liquid is travelling at a given velocity in a given direction, a change in direction will more easily deflect the vapors than the entrained drops. Due to their momentum the drops will not pass around a corner, and if once they come in contact with a wet surface, they coalesce with the surface film and are not easily picked up again by the vapor. In other words, rational entrainment separators should be designed on the basis of moving the mixture at a high velocity around a sharp corner. Screens or baffles are much less apt to be

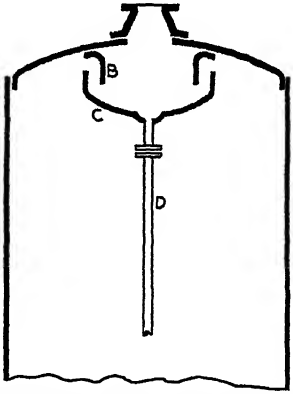

FIG. 88—Incorrect entrainment trap

effective; and separators which involve a drop in velocity are apt to be useless.

The actual constructions suggested for steam separators are so numerous as to be beyond even the briefest mention in this work. Schroeder[39] describes several hundred constructions from German patents, and the United States patents on these devices would run to several hundred more. Further, practically every description of an evaporator body will be found to include some type of baffles or other devices to prevent entrainment.

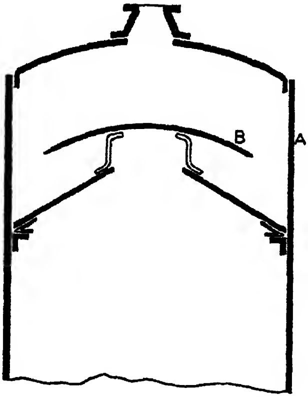

Fig. 89.—Incorrect entrainment trap.

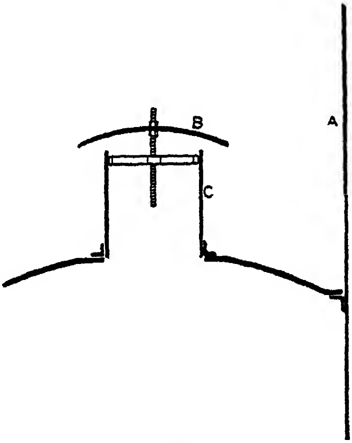

Fig. 90.—Heckmann entrainment separator.

The statement above, that a satisfactory entrainment separator involves turning a sharp corner at a high velocity, is not quite sufficient Schroeder[40] cites two very unsatisfactory constructions which apparently fulfil the above conditions. In Figure 88 the separator is formed of a tube B and a bell C. D is the return pipe for liquid Here such entrained liquid as was projected against the tube B, would have to drip off the lower edge into vapors of high velocity, and thus be carried away again. In Figure 89 the vapors, in passing out under plate B, were supposed to cause the liquid to impinge on its under side. The space between the plate and the wall of the evaporator A is so small that vapor velocities here were high, and drops were picked up from the edge of plate B. On the other hand, Figure 90 shows a very similar separator used by Heckmann in Germany. Here the space around plate B is large, the velocity between neck C and plate B high, and the velocity between plate B and shell A low. This results in liquid being caught on plate B, hurled sidewise from the

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[39] "Die Schaumabscheider," 1918, 160 pp.

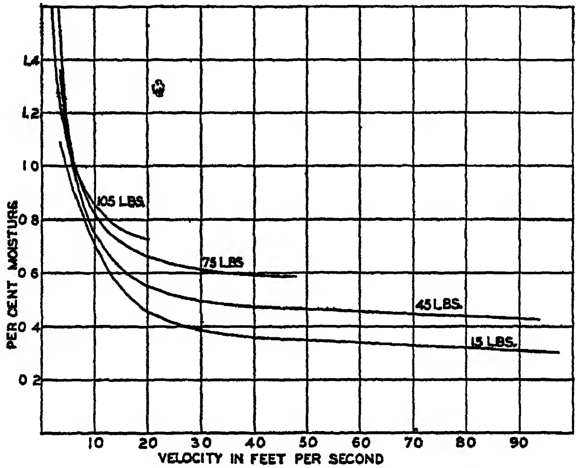

Fig 91—Tests of steam separator of Fig 92.

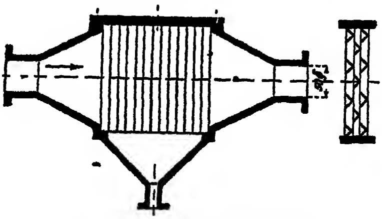

Fig 92.—Steam separator.

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[40] *Loc. cit.*, p. 11.

edge of B through a stream of low-velocity vapor, to impinge on shell A. The deflector on a basket type vertical acts in exactly the same way, but is much simpler. This is better than Figures 88 and 89, but a type in which the liquid which has been collected on a metal surface did not again drip through a stream of vapor would be better still. Hence the criterion of a good separator must be amplified by adding that liquid once separated should not drip through high-velocity vapor.

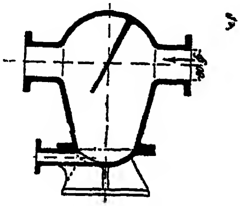

FIG. 93.—Steam separator.

The common fallacy that an entrainment separator should slow down the current of vapor is well refuted by the curves of Figure 91, quoted by Schroeder from Sendtner[41] on a separator of the type shown in Figure 92. The curves show the per cent moisture in steam after passing the separator at various velocities and at various pressures. The curves show how much better the action is at high velocities. They also show the effect of the density of the steam, confirming conclusion (2) from Stokes' law. A separator shown in Figure

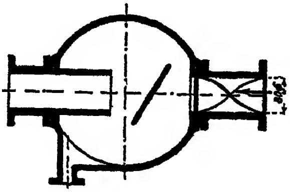

FIG. 94.—Steam separator.

93 gave similar curves, but a little lower moisture for the same velocities, and the separator of Figure 94 gave still better results. These are the only quantitative data on the performance of entrainment separators that have come to the writer's notice. Kerr[42] tested a separator of the type shown in Figure 95, in which separation is accomplished by centrifugal action. Here the insertion of twisters in the inlet of the separator gives the mixture a whirling motion. The greater mass of the entrained particles causes them to be thrown out into the separator while the outlet nipple takes the core of dry vapors.

[41] *Mitt. Forschungsarbeiten,* No. 98, 99 (1911).
[42] Bull. La. Agr. Exp. Sta. No. 138; *Met. Chem. Eng., 11,* 338 (1913).

Kerr passed the vapors leaving the separator to a surface condenser and tested the condensate for entrainment. The separation of entrained liquid was practically complete; his condensate showed no sugar in all but one test, and only a slight trace in that one. He found that the entrainment (measured by the weight of sugar in the material removed from the receiver as compared to the known concentration of sugar in the liquid boiled) varied with juice level, being greater at high levels. Two-inch tubes 48 inches long could only be run at about half the rating possible with 1¾-inch tubes 24 inches long without causing more entrainment with the long tubes than the short ones.

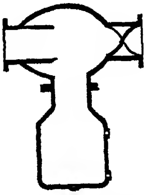

FIG 95—Steam separator.

Another type of separator involving the centrifugal principle is the one regularly used on the Kestner.[43]

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[43] Qualitative discussions of a few types of separators may be found, for instance, Claassen, *Centr Zuckerind*, *27*, 500-2 (1919); Helmer, *La Planter*, *62*, 107 (1919), *63*, 350 (1919), Van Ham, *Arch Suikerind*, *28*, 481-500 (1920); Haddon, *Rev. Agr. Maurice*, *1*, 274-5 (1922); Fries, *et al*, *Int. Sug. J*, *25*, 591-3 (1924)

Chapter 12.

Evaporator Auxiliaries.

In the field of auxiliary equipment for evaporators (condensers, vacuum pumps, traps, regulators), one finds hundreds of patents and dozens of descriptive articles, but an amazing lack of quantitative data. It would seem that such devices are left in most plants to the master mechanic, and technical supervision of their performance is rare. Consequently, this chapter will be mainly descriptive, although a little theory can be adduced as showing the relative importance of comparable devices.

CONDENSERS.

Condensers may be classified from several points of view, thus giving subdivisions that are co-ordinate rather than subordinate. The major division is into surface and jet condensers. The further subdivisions are applied in practice to jet condensers, although there is no reason why similar distinctions might not be made with surface condensers. A fair outline of the terms met may be made as follows:

Surface condensers
Jet condensers

Parallel-current	wet	Barometric
Counter-current	dry	Low-level

Definitions.

A *surface* condenser is one in which the vapor to be condensed and the cooling water are separated by metal surfaces; a *jet* condenser is one in which they are mixed. A *parallel-current* condenser is one in which the non-condensed gases are removed in equilibrium with the hottest cooling waters; a *counter-current* condenser is one in which they are in equilibrium with the coldest water. A *dry* condenser is one in which the non-condensed gases and the cooling water are removed by separate pumps; a *wet* condenser is one in which they are removed together. A *barometric* condenser is one in which the condenser is placed high enough to remove the hot water by gravity; a *low-level* condenser is one in which the cooling water has to be removed mechanically. Theoretically any combination under the above classification is possible; actually certain particular combinations are commonest. Thus, a parallel-current condenser is almost always wet, a counter-current condenser is usually dry.

Surface vs. Jet Condensers.

In a jet condenser the water used for cooling comes in contact with the vapors to be condensed, therefore the condensed water is mixed with the cooling water. The cooling water will be 20 to 50 times the weight of steam condensed, and is usually saturated with air. In a surface condenser this is not under vacuum; in a jet condenser this air is liberated when the water is introduced into the condenser.

In evaporator work extremely high vacua are not necessary, hence the lower vacuum due to air in the cooling water is not a disadvantage. Even in a single effect, the condensate from the heating surface is equal to the steam condensed and therefore the loss of condensed vapor in a jet condenser is no disadvantage Hence surface condensers are generally limited to power plants, where the highest possible vacuum is desired, and the condensate must be returned uncontaminated to be used as boiler feed. It is obvious that if the vapor from an evaporator is to be saved after condensation, a surface condenser must be used, but these cases are the exception. When it is recalled that the surface condenser is by far the more expensive type, its application to evaporator work is seen to be very limited.

With the growth of large central station turbines and the desire for the highest possible vacuum in their operation, many improvements have been made in surface condensers in recent years. The imposing list of patents in this field is sufficient indication of the interest it holds for the power engineer. These are of no significance so far as evaporator work is concerned, and will not be mentioned here. In the cases where surface condensers must be used on evaporators they are no different in principle from tubular heaters, and may be designed according to Chapter 8. The refinements mentioned above are of importance only after one has passed 28 to 28.5-inch vacuum—for 26-inch vacuum they are not significant.

Jet Condensers.

A simple type of parallel-current jet condenser is shown in Figure 96. Water enters at A, vapor at B. C is a cone for spraying the water into the body of the condenser, and water and air together leave at D. This is about the simplest possible type. Since there is no special device for removing air, the condenser outlet must be connected to a vacuum pump, which obviously must pump out both air and water. It is therefore a wet, parallel-current, low-level condenser.

There are many such condensers on the market, of a great variety of constructions, but all of the same type. The water may be distributed by perforated plates or a perforated pipe or by ring-shaped nozzles, but these are unimportant details. Many hundreds of such condensers are operating on small and medium sized evaporators, somewhat wasteful of water, but producing sufficient vacuum without excessive air pump capacities.

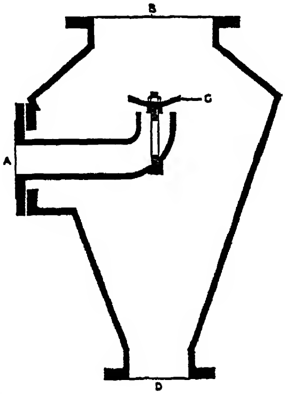
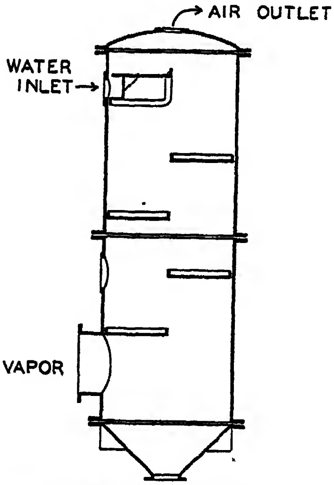


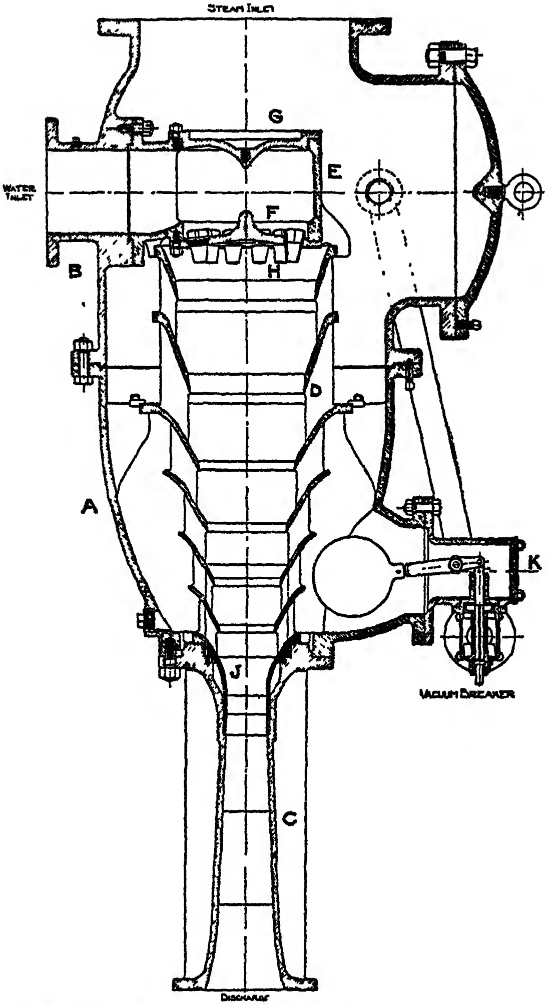
FIG. 96.—Parallel-current jet condenser.



Courtesy Swenson Evaporator Co.

FIG. 97.—Counter-current dry condenser.

A typical counter-current condenser is shown in Figure 97. The water cascades from shelf to shelf and leaves at the bottom. The vapor rises through the sheets of water, and the final non-condensable gas leaves at the top This is evidently a dry counter-current condenser, and it may be barometric or low-level according to circumstances


Steam Inlet
G
E
Water Inlet
F
B
H
D
A
K
J
Vacuum Breaker
C
Discharge

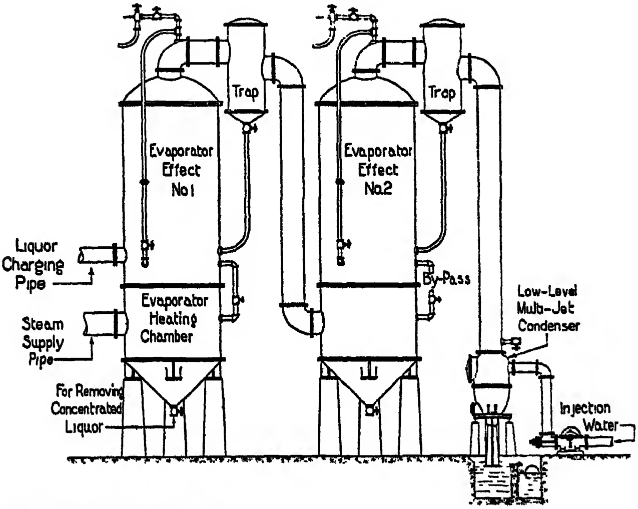
Courtesy Schutte and Koerting Co.

FIG. 98.—Ejector condenser

Figure 98 shows a condenser in which two different features are introduced. The water is introduced through nozzles H at fairly high pressures Diffusion vanes D are introduced to guide the vapors into contact with the jets without shock. The throat is constricted into a

Venturi tube at J, so that if sufficient water be used, its velocity at the narrowest section may be high enough to entrain the air and remove it. This type of condenser, therefore, need not be barometric in order to remove both air and water without a pump—provided the velocity head of the water at the constriction is greater than the static head corresponding to the vacuum. The constricted throat and automatic discharge of air may be used without the water jet feature. Figure 99 shows such a condenser connected to an evaporator.

There are hundreds of modifications of the internal constructions of these condensers, especially the simple counter-current ones. Trays, perforations, baffles, rotating sprays—every conceivable device, and every conceivable combination of devices, has been employed in mixing water and vapor. In spite of this development, the simplest constructions are still found to give perfectly satisfactory results. A list of patents bearing on jet condensers will be found at the end of this chapter.


Trap
Trap
Evaporator
Effect
No1
Evaporator
Effect
No2
Liquor
Charging
Pipe
Steam
Supply
Pipe
Evaporator
Heating
Chamber
By-Pass
Low-Level
Multi-Jet
Condenser
For Removing
Concentrated
Liquor
Injection
Water

Courtesy Schutte-Koerting Co.

Fig. 99.—Application of ejector condenser.

Parallel versus Counter-current Operation.

The advantages of counter-current operation are not immediately apparent, and are often misunderstood.

Let W = weight of steam condensed in pounds per hour.
P = total pressure in condenser.
a_0 = partial pressure of air in vapors entering condenser.

d_0 = partial pressure of water in vapors entering condenser.

t_0 = temperature of vapors entering condenser.

t_1 = inlet temperature of cooling water.

t_p = outlet temperature of cooling water in parallel-current operation.

t_c = outlet temperature of cooling water in counter-current operation.

α = elevation in temperature over t_1 of exit vapor in counter-current operation.

A = cubic feet air per hour passing to condenser, measured at a_0 and t_0.

V_p = air pump displacement in cubic feet per minute in parallel-current operation.

V_c = air pump displacement in cubic feet per minute in counter-current operation.

Q = pounds cooling water used per hour.

n_p = pounds cooling water used per pound steam in parallel-current operation.

n_c = pounds cooling water used per pound steam in counter-current operation.

L = latent heat of vaporization of steam at t_0.

$d_1, d_p, d_c, d_{1+\alpha}$ = vapor pressures of water corresponding to $t_1, t_p, t_c, t_{1+\alpha}$, respectively.

a_p, a_c = partial pressure of air at exit of condenser in parallel- and counter-current operation respectively.

It is to be understood that throughout this discussion "air" refers to any non-condensable gas. It will be assumed that the cross-section of all passages in the condenser is such that there are no pressure drops, and the total pressure in the condenser is P at all points.

It is obvious that in a mixture of gases we may think of each gas occupying the whole of the volume at a pressure less than the total, or occupying a part of the volume at the total pressure. Thus, in a cubic foot of steam-air mixture at 100 mm. total pressure, if the air forms 1 per cent of the total by volume, we may consider the cubic foot as consisting of

0.99 cubic foot steam at 100 mm.
0.01 " " air " " "

1.00 cubic foot mixture at 100 mm.

or (and this is the better way in condenser calculations)—

1.00 cubic foot steam at 99 mm.
1.00 " " air " 1 mm.

1.00 cubic foot mixture at 100 mm.

The temperature of the mixture corresponds to the *partial pressure* of the steam—not the total pressure. The above mixture would have a temperature of 124.5° F., corresponding to 99 mm.; not 124.9, corresponding to 100 mm.

Then, in the vapors entering the condenser from the evaporator

$$P = a_o + d_o \qquad (88)$$

In parallel-current operation, the air leaves at the temperature of the hottest water; so that

$$P = a_p + d_p \qquad (89)$$

The volume of air present, in cubic feet per hour, is

$$A \times \frac{a_o}{a_p} \times \frac{t_p + 460}{t_o + 460}$$

and

$$V_p = \frac{A(P - d_o)\ (t_p + 460)}{(P - d_p)\ (t_o + 460) \times 60} \qquad (90)$$

In counter-current operation the air cannot be cooled quite as far as t_1, but leaves at some temperature slightly above, such as t_{1+a}. The volume of air present is therefore

$$A \times \frac{a_o}{a_c} \times \frac{t_{1+a} + 460}{t_o + 460}$$

and

$$V_o = \frac{A(P - d_o)\ (t_{1+a} + 460)}{(P - d_{1+a})\ (t_o + 460) \times 60} \qquad (91)$$

Since t_{1+a} is always lower than t_p, the numerator of equation (91) will always be smaller than that of equation (90). Also, d_{1+a} will always be smaller than d_p, hence the denominator of equation (91) will always be larger than that of equation (90). Hence, the first advantage of counter-current condensers is that the volume of air to be removed, and consequently the air pump displacement, will always be less than with parallel-current operation for the same vacuum.

From definitions

$$n = \frac{Q}{W} \quad \text{and} \quad Q = nW$$

The heat entering the condenser in parallel-current operation is

$$WL + W(t_o - 32) + Q(t_1 - 32) \qquad (92)$$

The heat leaving the condenser is

$$W(t_p - 32) + Q(t_p - 32) \qquad (93)$$

Equating these and solving for Q we have

$$Q = \frac{W\,[L + (t_o - t_p)]}{t_p - t_1} \qquad (94)$$

and

$$n_p = \frac{L + (t_o - t_p)}{t_p - t_1} \tag{95}$$

In the same way, for counter-current operation

$$n_c = \frac{L + (t_o - t_c)}{t_c - t_1} \tag{96}$$

In equations (95) and (96), L is so much larger than the other terms in the numerator that any change here in t_p or t_c is of no significance. L varies through very narrow limits; hence it is the denominator which mainly affects the value of n This denominator is usually rather small (30 to 70° F) and hence a change of a few degrees in t_p or t_c will change n considerably.

If we compare formulas (95) and (96) with (90) and (91), we see that in counter-current operation d_c (which depends on t_c) does not enter into the formula for the volume of air to be removed. We may therefore increase t_c as far as we please, up to t_o as a limit, without in any way affecting the volume of gas to be removed, and hence (if the displacement of the air pump be fixed) without affecting the vacuum produced. In parallel current operation, on the other hand, if we increase t_p in an attempt to decrease n_p, we also increase d_p. But the denominator of equation (90) is a constant times $P - d_p$. As t_p approaches t_o, d_p approaches d_o. In ordinary operation a_o is very small and p_o is almost equal to P Therefore, as d_p approaches d^o, $(P - d_p)$ approaches zero and the volume of air to be removed increases. If we made the exit temperature of the cooling water equal to the steam temperature, as we did in counter-current operation, V_p would be infinite; in other words, the condenser would be doing nothing and the vacuum pump would have to pump the same mixture of steam and air as was coming from the evaporator. Hence the principal difference between parallel- and counter-current operation is that in counter-current operation the exit water may be as hot as the steam, while in parallel-current operation the exit water must always be considerably colder than the steam. Hence *a parallel-current condenser always uses more water than a counter-current condenser.*

Problem.

A vacuum of 26 inches referred to a 30-inch barometer is to be maintained in an evaporator. The air comes to the condenser in amounts such that the vapors carry 0 2 per cent air by volume. The last effect is producing 2000 pounds steam per hour. Cooling water is available at 60° F. How much water will be needed, and what should be the displacement of the air pump, in counter-current operation? Assume $\alpha = 5°$.

An absolute pressure of 4 inches $= 4 \times 25.4 = 101\ 6$ mm. The partial pressure of the air is then 0.2 mm. and hence

$$\begin{aligned} P_0 &= 101.4 \text{ mm.} \\ a_0 &= \underline{\ \ 0.2} \\ P &= 101.6 \end{aligned}$$

$$t_0 = 125.4° \text{ F.}$$

The specific volume of steam at this pressure is 176.7 cubic feet per pound, hence the volume of steam entering is $2000 \times 176.7 = 353{,}400$ cubic feet at 101.4 mm. The air therefore occupies 353,400 cubic feet at a pressure of 0.2 mm.

From equation (91)

$$V_c = \frac{A(P - d_0)\,(t_{1+a} + 460)}{(P - d_{1+a})\,(t_0 + 460) \times 60}$$

$$t_{1+a} = 65°, \text{ hence } d_{1+a} = 15.8 \text{ mm.}$$

Substituting in (91)

$$V_c = \frac{353{,}400\,(101.6 - 101.4)\,(60 + 5 + 460)}{(101.6 - 15.8)\,(125.4 + 460) \times 60} = 12.31 \text{ cubic feet per minute.}$$

From equation (96)

$$n_c = \frac{L + (t_0 - t_c)}{t_c - t_1} = \frac{1021 + (125 - 125)}{125 - 60} = 15.7 \text{ pounds } H_2O \text{ per pound steam.}$$

$$Q = n_c W = 2000 \times 15.7 = 31{,}400 \text{ pounds water per hour.}$$

To compare this with parallel-current operation, let us determine V_p and n_p where t_p is (a) 115° and (b) 100°, respectively.

For $t_p = 115°$, $d_p = 75.8$ mm.; and, from equations (90) and (95)

$$V_p = \frac{A(P - d_0)\,(t_p + 460)}{(P - d_p)\,(t_0 + 460) \times 60} = \frac{353{,}400 \times 0.2 \times 575}{25.8 \times 585.4 \times 60} = 44.8 \text{ cubic feet per minute.}$$

$$n_p = \frac{L + (t_0 - t_p)}{t_p - t_1} = \frac{1021 + 10}{115 - 60} = 18.7 \text{ pounds water per pound steam.}$$

Similarly, where $t_p = 100°$, $d_p = 50.7$, $V_p = 22.14$ cubic feet and $n_p = 26.2$ pounds.

Hence, in this case, in changing from counter-current operation to parallel-current operation at 115°, the air pump capacity must be increased nearly four times, while the water consumption increases about 20 per cent. If the parallel-current condenser be operated at 100° instead of 115°, the air pump displacement need not be over twice that of the counter-current condenser, but 67 per cent more water will be used than in counter-current operation It is obvious that the differences in water consumption will be greater as the inlet water temperature is higher The design of the counter-current condenser may be such that if the water be reduced to the theoretical limit, there will not be water surface enough to condense all the steam, and hence t_c may not equal t_0. It can be made nearly equal to t_0 in most properly designed condensers, unless the condenser is operated on volumes of steam much smaller than those for which it was designed.

Air Entering the Condenser.

This comes from four sources:

1 Air contained in the steam used for heating.
2. Air dissolved in the feed liquor.
3. Air entering through leaks in the apparatus.
4. Air dissolved in the condenser water

If the steam space of the first effect is above atmospheric pressure, it may be vented into the room, and hence item (1) does not affect the condenser. If the first effect steam space is vented to the vapor space, all this air, together with all of item (2), passes to the condenser. Items (3) and (4) are obvious. The comparisons made in the problems above are, therefore, more favorable to the parallel-current condenser than they should be, for the larger amount of water used in a parallel-current condenser means that it will always have more air to remove than the counter-current condenser, whereas the problems were worked on the basis of equal amounts of air.

The actual quantities to be expected are not known. Very few determinations of amounts of air in condensers have been made. Weiss[1] suggests using the following factors:

(4) Air in cooling water—2 per cent of the cooling water by volume, measured at 60° F. and 30 inches Hg.
(1), (2), (3)—Air from the evaporator—0.5 per cent of the steam entering the condenser by volume.

This latter value is based upon six tests in beet sugar factories in which a quadruple effect evaporator and a single effect strike pan were connected to a single condenser. This figure is probably high as compared to general practice—first, because beet sugar juices evolve considerable ammonia on boiling; second, because there are here five bodies connected to the one condenser.

Corner[2] suggests the following figures, based on sugar evaporators:

Air entering in cooling water, 0.1 pound per 1000 pounds water.
Air entering in vapors from the evaporator—

---

[1] "Kondensation," 2nd Ed., Berlin, 1910, pp. 28-38.
[2] *Eng*, *109*, 74-5 (1920).

Single effect	0.28 per cent by volume
Double effect	0.29
Triple effect	0 30
Quadruple effect	0.37
Quintuple effect	0.39

Coxon[3] suggests 0.25 to 0.35 per cent on the vapors by volume, for cane sugar evaporators. Corner's figure for air in the cooling water corresponds to 0.8 per cent by volume as compared to Weiss's figure of 2 per cent. Corner says his figures are based on experience. Weiss bases his figure on the solubility of air in water, and admits that the cooling water is not always saturated, and that all the air is not removed from the water in the condenser. It would seem that from 1 per cent to 2 per cent of the volume of cooling water would be a safe figure for the air from this source. Weiss is higher than Corner on air from the evaporator, but it is not possible to judge between these figures. Where the solution does not evolve any gas other than dissolved air, and where an evaporator of three effects or less is used, 0.20 per cent by volume should be safe. Each case must be judged on its own merits but a real quantitative estimate is difficult to make. A little carelessness in erection would easily double any of the above figures.

We may now proceed to formulate A more exactly, and apply the above values in a more rigorous solution of the previous problems. The method of procedure depends on which condition is fixed, but in the following discussion we shall assume that t_o (the temperature of the vapors entering the condenser) is a fixed condition. The steps will then be (a) determine the volume of the steam, knowing p_o (which is fixed by t_o); (b) knowing the per cent air by volume in the steam, determine a_o; (c) determine Q, and hence the volume of air at 60° F. and 30 inches Hg entering the condenser from the cooling water; (d) determine the pressure of this air when expanded to the volume and temperature of the steam. If we call this pressure a'_o, the total pressure in the condenser, P, will be $t_o + a_o + a'_o$. If we denote the volume of air entering with the water, measured at 50° F. and 30 inches Hg by A_w; and its volume at condenser temperatures and pressures by A′, we may write

$$A' = A_w \times \frac{t_o + 460}{t_1 + 460} \times \frac{762}{a'_o} \qquad (97)$$

from which, having A′ known,

$$a'_o = \frac{A_w}{A'} \times \frac{762\,(t_o + 460)}{t_1 + 460} \qquad (98)$$

We may now repeat Problem 1 with the following additions: Assume the vapors from the evaporator carry 0.2 per cent air by volume, and that the cooling water carries 1.5 per cent by volume. Assume that a temperature of 125.4° F. is to be maintained. What must be the air-pump displacement for (a) counter-current operation, (b) parallel-current operation with $t_p = 115°$ F., (c) counter-current operation with $t_p = 100°$ F.?

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[3] *Int. Sug. J., 19,* 19 (1917).

These additions to the problem do not affect the quantities of cooling water determined in the previous solution, and we may use $n_c = 15.7$ pounds, n_p (for 115°) $= 18.7$ pounds, and n_p (for 100°) $= 26.2$ pounds. For case (a):

$$Q = n_c W = 15.7 \times 2000 = 31{,}400 \text{ pounds cooling water per hour.}$$

$$\frac{31{,}400}{62.4} \times 0.015 = 7.53 \text{ cubic feet free air per hour.}$$

A′ is the same as the steam volume, which is 353,400 cubic feet per hour.

$$a'_o = \frac{A_w}{A'} \times \frac{762\,(t_o + 460)}{t_1 + 460} = \frac{7.53 \times 762 \times (125.4 + 460)}{353{,}400 \times (60 + 460)} = 0.018 \text{ mm}$$

Since $t = 125.4°$, $p = 101.4$ mm. The mixture contains 0.2 per cent air by volume, hence $a_o = P_o \times \frac{0.2}{98} = 0.21$ mm.

$$\begin{aligned} P_o &= 101.4 \text{ mm.} \\ a_o &= 0.21 \text{ "} \\ a'_o &= \underline{0.02} \text{ "} \\ P &= 101.63 \text{ mm.} \end{aligned}$$

In using equation (91) we must now use $(a_o + a'_o)$ in place of a_o in the equation as written. Hence

$$V_c = \frac{353{,}400 \times 0.23 \times 525}{85.8 \times 585.4 \times 60} = 14.16 \text{ cubic feet per minute}$$

In the previous solution we had 12.31 cubic feet, hence a consideration of air in the water increases the vacuum pump displacement by 15 per cent. This is the limiting case, as this particular method uses the least possible water under the conditions of the problem.

In a similar way, for case (b) we find that $A_w = 8.79$ cubic feet, $a'_o = 0.021$ mm., and $V_p = 51.4$ cubic feet (previously 44.8, an increase of 15 per cent); while for case (c) $A_w = 12.6$ cubic feet, $a'_o = 0.030$ mm., and $V_p = 26.6$ cubic feet, an increase of 20 per cent. Obviously, as n increases, whether parallel- or counter-current, V_p or V_c must also increase; and the increase will be relatively more rapid as n is larger and the amount of air introduced in the water is more important as compared to that in the vapors.

From the above formulas and problems, we may draw some qualitative conclusions regarding the effect on condenser operation of changing the amount of water used. If a counter-current condenser is operated with an amount of water larger than the optimum, the temperature of the air to be removed (and hence its content of water vapor also) is not affected. The amount of air to be removed will be larger than necessary. If the supply of water be decreased till the exit water practically reaches steam temperature, the only effect is to reduce the amount of air brought in by the cooling water. Air from the evaporator proper will not be affected in volume, temperature, or water-vapor content. Hence in a counter-current condenser, the highest vacuum for a given air-pump displacement (or the lowest air-pump speed for a given vacuum) is reached when the least water is used.

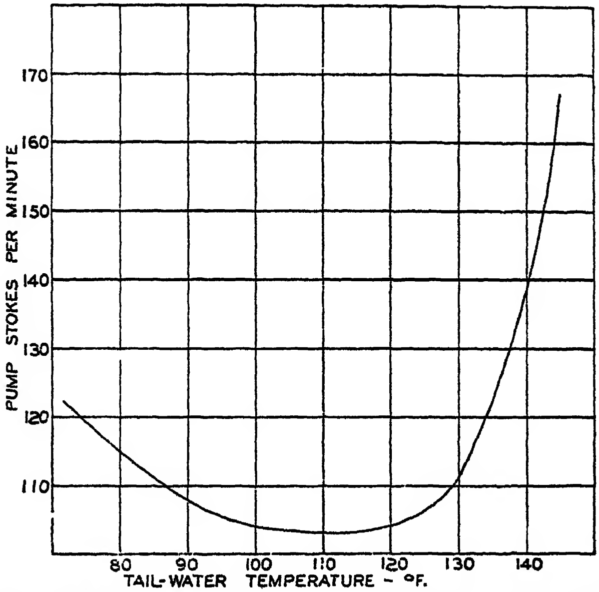


FIG. 100.—Effect of amount of cooling water on air pump displacement.

In a parallel-current condenser, on the other hand, the relation is not so simple. If we imagine a very small amount of water used, the air-pump displacement needed to maintain a given vacuum is very large, because of the high partial pressure of water in the discharge mixture. As we increase the supply of water to the condenser, the air entering the condenser increases, but the temperature of the air-steam mixture to be removed decreases. This decreases its volume by decreasing the partial pressure of water vapor in it. The result is a lowering of the air-pump displacement necessary to maintain the vacuum.

As we keep on increasing the supply of water, equal increments of water bring in equal amounts of air, but do not lower the exit water temperature by equal amounts. Hence, after a time, we will reach the point where air brought in by the additional water more than makes up for the cooling caused by this additional water, and the air-pump displacement needed increases again.

Figure 100 shows some tests run in the author's laboratory on a single effect vertical tube evaporator, connected to a parallel-current wet condenser like Figure 96. Steam to the evaporator was held constant at 209.5° F., cold water fed to the condenser was varied, and by throttling the vacuum pump the vacuum was held approximately constant. On the branch of the curve corresponding to high tail-water temperatures, and around the minimum, the absolute pressure in the evaporator varied from 10.8 inches to 11.6 inches (165 to 168° F.). With large amounts of water the pump could not be run fast enough to maintain the vacuum and on this branch of the curve the absolute pressure rose to 12.3 inches (171° F.). Hence, to have obtained a constant vacuum the pump should have been run faster, and this branch of the curve does not rise steeply enough.

The position of the minimum of this curve is not of general significance (i.e., all parallel-current condensers will not call for the minimum vacuum pump capacity at a tail-water temperature 45 to 50° F. below the temperature of the steam condensed). It does indicate that there are optimum operating conditions which must be determined for each condenser.

VACUUM PUMPS.

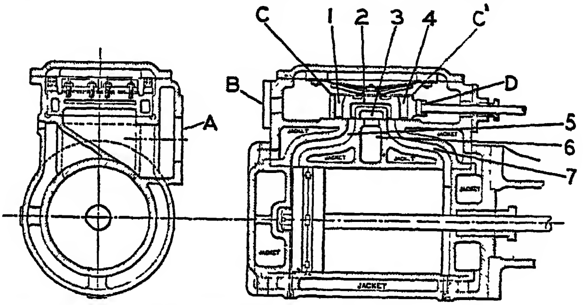
Vacuum pumps may be classified as follows:

- Reciprocating pumps
    - Disc valves
    - Slide valves
        - Simple
        - Equalizing
    - Rotary valves
    - Spring valves
- Rotary pumps
- Jets or ejectors

Reciprocating Pumps.

The simplest vacuum pump is the ordinary wet vacuum pump used for removing both air and water from a parallel-current wet condenser. Such pumps are essentially ordinary tank pumps with the usual hard rubber valves, the only difference from tank pumps being that the water end castings may be a little lighter and the discharge air chamber is usually omitted. Such a pump may be used as a dry vacuum pump (i.e , to remove air from a dry condenser, as defined on p. 215), but in this case a little cold water must be fed into the air cylinder to seal the valves and help fill the clearance spaces. If this water is as cold as the inlet water to the condenser, the operation is still counter-current; if it is as warm as the exit water to the condenser, all the advantages of counter-current operation are lost and the condenser becomes parallel-current.

In the case of dry condensers a really dry pump is often desirable, as higher speeds, lower clearances, and more positive-acting valves are then possible. Such pumps may have ordinary D-slide valves, piston valves, rotary valves of the Corliss type, or spring-loaded discharge valves. In such cases, the clearance in the air cylinder limits the vacuum which may be obtained. If, for instance, the clearance is 10 per cent of the displacement, when the vacuum in the condenser reaches 0.1 atmosphere, the air drawn into the cylinder on the suction stroke is compressed to fill the clearance at just 1 atmosphere on the discharge stroke, and hence no air can be discharged. Since many evaporators do not operate at pressures less than 4 inches absolute (26 inches vacuum on a 30-inch barometer), and since this calls for a dead space in the air cylinder of 13 per cent of the displacement or less, special valves are not often needed. There are dry vacuum pumps on the market with simple Corliss or D-slide intake valves, and spring discharge valves, whose clearances are small enough to give 27 inches vacuum.



Courtesy Worthington Pump and Machinery Corporation

FIG. 101 —Slide valve air pump with equalizing port.

With older constructions, an equalizing port was introduced into the valve. Such a pump is shown in Figure 101. A is the inlet, connecting the port 6 in the cylinder body. The slide valve D has an inlet port 3, discharge ports 1 and 4, and a special equalizing port 2. The discharge valves proper, C and C′, are thin strips of metal hinged in the middle, arranged to open at pressures slightly above atmospheric. Air escaping from these valves C finally leaves at B.

While the main piston was travelling from right to left, just before reaching the position shown, the slide valve D was a little to the right of its present position. Port 3 then connected ports 6 and 7, and port 1 registered with port 5. As air was compressed in the left-hand end of the cylinder, when it reached atmospheric pressure it would lift valve C and escape. At the end of the stroke, therefore, the clearance spaces would be filled with air at atmospheric pressure If the volume of the dead space was 1/nth of the displacement, and the vacuum at A was 1/nth of an atmosphere, on the return stroke of the piston this air would expand to fill the cylinder at the pressure of the condenser and no further lowering of the vacuum could occur.

This is remedied by the introduction of equalizing port 2. At the end of the stroke, when the valve D was in a position to the right of that shown, the right-hand end of the cylinder contained air at condenser pressures. As the valve moved from this position to the one shown, ports 1, 4, and 6 were closed, and for an instant port 2 connected ports 5 and 7. This allowed the air in the dead space to the left of the main piston to expand into the space at the right of the piston, raising the pressure of the air there *above* 1/nth of an atmosphere. The final position of valve D is to the left of the one shown, port 3 connecting ports 5 and 6, and port 4 registering with port 7. Since the air to the right of the piston is now above 1/nth of an atmosphere, on being compressed into the clearance space it will be above one atmosphere, and hence some can be discharged. Thus the presence of an equalizing port such as used here makes possible a higher vacuum than the ratio of the clearance to the displacement would indicate.

Such equalizing ports may be incorporated in piston type valves or Corliss valves, or may be included in a second independent slide-valve. With modern constructions such devices are no longer necessary, and the construction may be considered obsolete. There are still many such pumps in service, but no new installation need consider them.

Rotary or Hurling Pumps.

Rotary or hurling pumps are more used in power plant work than in evaporator work. At one time they were much discussed, but they seem to be superseded now by the jet ejectors to be mentioned in the next section. These pumps consisted of a standard double suction volute centrifugal pump, pumping cold water. From the discharge of the impeller of such a pump there issues a flat sheet of water, not quite solid because of the vanes in the impeller. Such a sheet of water then passed into a narrow set of passages where it acted like a water-jet ejector, carrying air from the condenser (which was introduced into the pump in an annular passage surrounding the impeller discharge) out through these passages to the volute, where air and water together were discharged into an open well. Since the air was already cool, the water was not heated and could be used again; and since this water could be used at t_1, the advantages of counter-current operation were not lost and the pump was, therefore, a "dry" pump.

The Rees Roturbo condenser is a similar device, but the sheet of water issuing from the pump impeller becomes the condenser water also, so that both condensation and air removal are combined in the same device.

Steam Jet Ejectors.

These devices are the outstanding development in power plant condenser practice in recent years.[4] They are not used in evaporator practice to any extent. One of the best known is the Radojet ejector, Figure 102. 58 is a connection for live steam at 110 pounds or over, S is the air suction, and D the discharge. Steam passes by valve 3 and chamber 4 to the upper set of nozzles 5 which entrain the air and compress it somewhat. This mixture of air and steam passes through chambers 8 and 9 to the second jet. This second jet consists of steam entering by chambers 10 and 11, and impinging on the disc 12. The resulting sheet of steam at high velocity entrains and discharges the air from the first nozzles through passage 14 and volute 15.

The first stage nozzles will compress air from 0.25 inch absolute pressure up to 4 to 5 inches absolute. Hence if a condenser need not work at vacua higher than 25-26 inches, the second stage only is needed. The apparatus is very compact, and its first cost is low, but unless the steam used in it can be used for heating feed, it must be wasted. A jet uses more steam than a reciprocating vacuum pump at vacuums of 26 inches or less. At higher vacuums the jet uses less steam than the reciprocating pump. The exhaust from a vacuum pump can be used in evaporation, but if the exhaust from this device were so used, the air would be returned to the evaporators in a closed system. Ejectors deserve more consideration in evaporator work than they have yet received. For small condensers, a common reciprocating wet vacuum pump, and for dry condensers, modern low-clearance reciprocating pumps with spring valves, still seem to be the most popular.

CONDENSATE REMOVAL.

Condensate may be removed from evaporator bodies by pumps or traps. In many respects pumps are considered more reliable and more positive in their action. Also, a condensate pump may be made oversize and allowed to remove air as well as water, which cannot be done with traps unless the steam space is above atmospheric pressure. Traps are, nevertheless, widely used on evaporators of all sizes.

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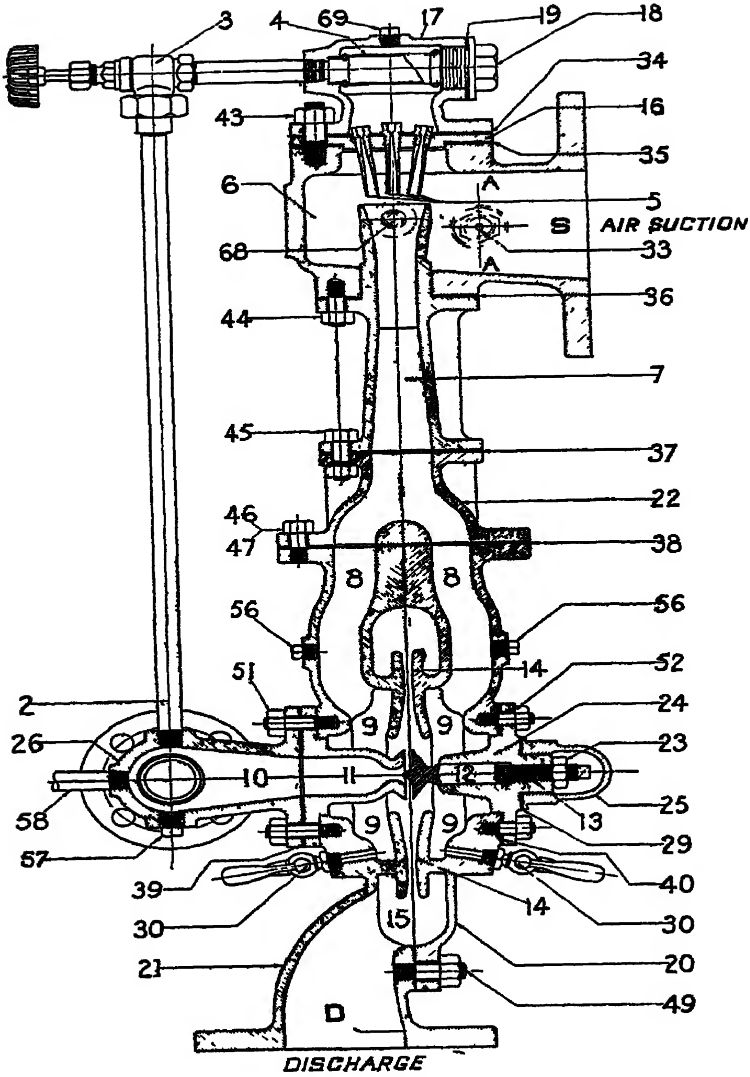
[4] See a long list of patents in recent years, mainly by Kothny and Suczek, but also by Bancel, Delas, Parsons, and various engineers of the Westinghouse Co.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D
DISCHARGE

Courtesy C H. Wheeler Mfg. Co.

FIG. 102.—Radojet air pump.

Traps.

Traps may be classified as follows:

Expansion
Float
Bucket
Tilt

We may also distinguish between return and non-return traps. A *return trap* is one which will discharge water to a space of higher pressure than the space it drains; a *non-return trap* can discharge only to a space of lower pressure than the space it drains. Expansion traps are always non-return; float and bucket traps are generally non-return, and tilt traps may be either.

There is a very large number of traps on the market, of a bewildering variety of designs. In the following paragraphs one trap of each type will be described, without implying anything except that it is typical. Many engineers go so far as to say that there is no such thing as a good trap. It is certain that any trap, to give satisfactory service, must have frequent attention by an experienced man.

Expansion Traps.

Figure 103 shows an expansion trap. To one end of the cartridge is attached a corrugated tube of thin metal, and the whole cartridge is filled with an oil and sealed. The valve is attached to the inner end of the corrugated tube. So long as water enters the lower connection of the trap body, the oil is cool, the corrugated tube is extended inward, the valve is open, and the water is discharged, provided that the space to which the trap is discharging is at a lower pressure than the space drained. When water has ceased to flow and steam enters the trap, the temperature of the oil rises and it expands, the corrugated tube is compressed, the valve closes, and steam flow is stopped. This type of trap cannot be made into a return trap.

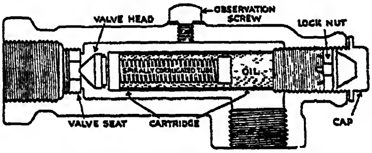


FIG. 103.—Expansion trap.

Bucket Traps.

Figure 104 shows one of many designs of bucket trap. Water enters the body of the trap as indicated, floats the bucket, and rises until it overflows into the bucket. When the bucket has received enough water, it sinks, pulls down on the valve stem, opens the valve, and thus allows water to blow out of the bucket. When the bucket is empty it floats, closes the valve, and the process is repeated. Float traps are similar in principle, a closed float rising, as water accumulates, and thereby opening a valve. The valve may be above the water level with a tube projecting into the water as in Figure 104; or the valve and its mechanism may be entirely below water level. The latter arrangement makes it a little easier for mud or scale to enter the valve.

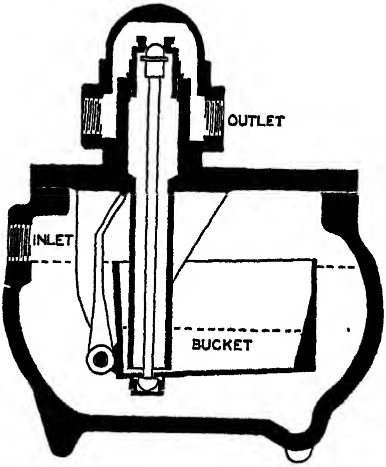
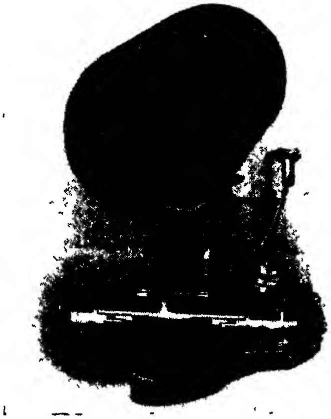


Fig. 104—Bucket trap.

Tilt Traps.

Tilt traps may be either return or non-return. The Moorehead non-return trap is shown in Figure 105. The receiver is supported on trunnions shown in section at the front, and held up by a lever and counter-weight. When in this position, the valve at the right is closed. Condensate drains into the trap until it overbalances the counter-weight, when the rear of the receiver falls, opening the valve and allowing condensate to flow through. In filling, condensate enters by the upright connection at the front; in dumping, it leaves by the rear connection, flowing forward and out through the trunnion to the right.

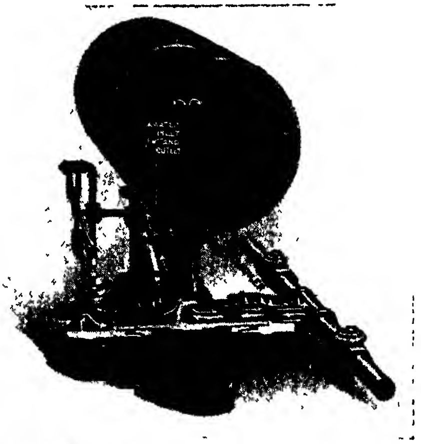
The Moorehead return trap is shown in Figure 106. Condensate enters by the connection to the right at the point marked "to trap." By definition, a return trap is one which will discharge to a space of higher pressure than the pressure in the space it drains. The connection marked "to boiler" is the discharge, and the two check valves shown are so placed that the pressure in the discharge space holds the check valve in the discharge line closed during filling.



Courtesy Moorehead Mfg. Co.

FIG. 105.—Non-return tilt trap.

Under the seat of the valve shown to the left is a connection provided with a valve just behind the one shown. This line is connected to the space to be drained or to a space of lower pressure. If, for instance, this trap were draining the steam chest of the last effect of an evaporator, this vent line would be connected to that steam chest, or to the vapor space of the same effect, or to the condenser. The



Courtesy Moorehead Mfg. Co.

FIG. 106.—Return tilt trap.

steam valve shown is closed during filling, and the vent valve behind it is open. This vent line connects through the left-hand trunnion with the vertical pipe at the front, and therefore serves to put the body of the trap under the same pressure as, or less than, the space being drained Condensate therefore flows in through the right-hand trunnion, to the back of the receiver, and fills it.

When the weight of condensate in the receiver is great enough to overbalance the counterweight (not shown) the back of the receiver drops, the trap rotating on the trunnions. This operates two levers which simultaneously close the vent valve and open the steam valve. The pressure in the receiver is now *greater* than in the space drained, the check valve in the feed line closes, the check valve in the discharge line opens, and the condensate is discharged. When the receiver is empty the counter-weight lifts it, thereby closing the steam valve and opening the vent valve, and the process is repeated.

If this trap is situated above the water level of a boiler, it will discharge condensate to the boiler directly. This is the reason for the term "return trap" and also the reason for labelling the discharge line "to boiler." A trap draining a vacuum evaporator and discharging to an open hot-well is just as much a return trap as in any other case.

If traps are used to drain evaporators or heating coils, which use steam at pressures above atmospheric, a pet cock on the trap body may be left open to vent non-condensed gases. If the steam space drained be at less than atmospheric pressure, there is no way of removing non-condensed gases by a trap, whether return or non-return; unless in the case of a return trap the vent be connected to the condenser. A method of removing air from the bottom of the steam space is most desirable, no matter what other air-vents may be provided. This accounts for the general disinclination to use traps on multiple-effect evaporators, except on the first body where the steam space is usually under pressure. Reciprocating pumps are usually preferred for the reasons mentioned above. If traps are used at all, they should be tilt traps or some other form of return traps, and vented to the condenser, not to the space they drain.

PATENTS ON JET CONDENSERS

Helander	785,613 (1905)	Fleming	887,081 (1908)
Helander	785,614 (1905)	Helander	887,634 (1908)
Tomlinson	789,945 (1905)	Helander	908,405 (1908)
Keilholtz	805,141 (1905)	Koerting	926,198 (1909)
Grace	809,679 (1906)	Petsche	947,845 (1910)
Dillon	813,486 (1906)	Schwanhausser	960,417 (1910)
Newhall	815,031 (1906)	Tomlinson	1,002,814 (1911)
Schutte	816,043 (1906)	Mueller	1,015,822 (1912)
Schwanhausser	824,312 (1906)	Trinks	1,028,156 (1912)
Heisler	825,831 (1906)	Trinks	1,028,157 (1912)
Koerting	847,010 (1907)	Tomlinson	1,036,031 (1912)
Roberts	857,675 (1907)	Beyer	1,041,954 (1912)
Schneider	864,925 (1907)	Norris	1,043,135 (1912)
Cramer	884,077 (1908)	Helander	1,048,532 (1912)

PATENTS ON JET CONDENSERS—*Continued*

Helander	1,068,430 (1913)	Horstmann	1,367,743 (1921)
Sleeper	1,115,980 (1914)	Engel	1,399,294 (1921)
Helander	1,144,197 (1915)	Morgan	1,457,788 (1923)
Helander	1,144,198 (1915)	Grace	1,474,358 (1923)
Woodard	1,153,001 (1915)	Scanes	1,483,989 (1924)
Ray	1,160,652 (1915)	Morgan	1,516,236 (1924)
Beyer	1,162,969 (1915)	Morgan	1,516,237 (1924)
Buss	1,179,369 (1916)	Wilson, *et al*	1,516,939 (1924)
Mesta and Trinks	1,224,310 (1917)	Bancel	1,524,281 (1925)
Helander	1,248,112 (1917)	Bancel	1,524,807 (1925)
Ehrhart	1,255,518 (1918)	Bancel	1,526,041 (1925)
Helander	1,319,647 (1919)		

Chapter 13.

Evaporators as Applied in Specific Industries.

This chapter is intended more as a collection of instances which may serve to illustrate principles from the preceding chapters, rather than as a manual for engineers already familiar with these industries. Further, devices and practices from one field may furnish ideas to engineers in totally different lines. Unfortunately, in many industries current practice is dictated by precedent or prejudice rather than by careful consideration of various possibilities; consequently not all the methods cited here are to be followed with similar materials.

SUGAR MANUFACTURE.

Beet Sugar.

Practically the whole history of the development of multiple effect evaporation may be found within this one industry. Although Rillieux's original evaporator was built for cane sugar, it was immediately taken up by the German and French beet industry; and from then on almost every known type of evaporator was first tried on beet juices. Further, the beet sugar industry has brought Rillieux's idea of the use of vapors from the evaporator to a higher degree of development than any other industry.

An exhaustive discussion of the evaporation of beet juices and the steam flow sheets of beet mills would be superfluous in this book because of the volume of material already published in standard works and technical journals. Consequently the present discussion will be brief, and more or less for the benefit of the engineer outside the industry.

The average beet juice when ready for the evaporator contains about 15 per cent solids, is clear and not very viscous, has a very slight elevation in boiling point, and contains a number of scale-forming constituents. It is evaporated to about 60 per cent total solids, at which point it is a viscous sirup, not yet saturated at last effect temperatures, and with a boiling point elevation of about 6° F. This thick juice is usually sent to storage tanks, hence the temperature at which it leaves the evaporator need not be high. It must later be boiled till the desired crystals are formed. In addition to the regular demands for steam for heating evaporator feed, for evaporating, and for boiling to grain, there are many steps in the process where the juice must be heated. Claassen[1] lists these items, as pounds of steam used per 100 pounds of water evaporated, as follows:

	Pounds
Raw juice heaters	10
Diffusion heaters	8
Carbonation heaters	7
Thin juice heaters	4
Thick juice heaters	1
Boiling to grain	12

The power demands of a sugar mill are such that the exhaust from the prime movers will not nearly suffice to operate these heaters and run the evaporators, if exhaust steam is used for heating.

Both vertical, horizontal, and film type evaporators are used, the vertical being most popular in Europe and in Cuba, the horizontal in this country. Ease of cleaning is the main argument for the vertical; but the scale is fairly easily removed from a horizontal by boiling with caustic followed by hydrochloric acid. Ease of tube renewals is the main argument for the horizontal. There has never been a discussion of the relative merits of the two types in which actual costs, overhead and operating, have been compared for similar installations on similar service.

Since the stronger juices are sensitive to high temperatures, feed is always forward, though this puts a severe handicap on the last effect, where the most viscous juice boils at the lowest temperature. The first effect should not boil at temperatures much over 240° F. (115° C.); though it is claimed that the short time of contact of the juice with the heating surface in high-speed evaporators like the Kestner makes possible boiling temperatures as high as 250° to 260° F. (120°-130° C.) in the first effect. This fact, together with the fact that large amounts of exhaust are available for evaporation, limits the temperature range through which sugar evaporators may operate.

The various heaters mentioned above are usually supplied with vapor from the first and second effects. Some heating is often done with vapor from the last effect. The use of vapor heaters is much more common in Germany than in this country. Many mills in this country withdraw vapors only from the first effect. Some use no vapor heaters at all, but do all their heating with live or exhaust steam. In Europe it is not uncommon to condense in vapor heaters 35 to 40 per cent of the total evaporation. The usual evaporator, both here and abroad, is a quadruple effect. A few quintuple effects are in operation, and a few triples. The triples are admittedly uneconomical under present conditions.

Vacuum Pans.

Boiling to grain is done in single effect and by batches, both because the material is extremely viscous toward the end of the operation, and because great care must be used to control crystal size during the boiling. This apparatus is known as the "vacuum pan."[2] Some American mills use pans with copper coils, some use pans exactly like a vertical tube evaporator The "express" construction (p 141) is common in Europe and is used to a certain extent here.

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[1] "Beet Sugar Manufacture," translated by Hall and Rolfe, Wiley, 1911, p. 334.

Because of the viscosity of this mass of crystals in a heavy mother liquor, there is a tendency to use large temperature drops, and in this country the usual practice is to operate the pan with live steam. Abroad, exhaust is largely used; and when this is the case, the heating surface of the pans must be larger than when live steam is used

Pre-evaporators.

Many of the older mills abroad had power units designed for low back pressures and therefore they could not carry as high boiling points in the first effect as the properties of the juice would permit. At the same time, some live steam was needed in the vacuum pans. The use of a reducing valve to step down live steam to exhaust pressures seemed wasteful The idea that an evaporator was essentially a reducing valve was adopted simultaneously by Pauly and by Greiner. Another effect was placed in front of the usual evaporators (i e, on the high-pressure side). This body was heated by live steam, and its vapors sent partly to the first effect and partly to the vacuum pans. Thin juice was fed first to this extra effect and then through the evaporators in series. This extra effect is known as a *pre-evaporator,* or a Pauly evaporator, or a Greiner-Pauly evaporator. In some cases ordinary vertical tube bodies are used, in some cases special high-speed evaporators like the Kestner. In a number of cases two pre-evaporators in series are used This makes essentially a sextuple effect evaporator; the juice goes through all six bodies in series, live steam is used to heat the first effect, exhaust is added to the steam space of the third, vapors are taken off the second for the vacuum pans, and vapors off the second, third, and fourth for heaters This system of one or two pre-evaporators is common in Germany but not often used in this country.

Pressure Evaporation.

The quantity of steam used for heating is so large that if it is supplied as vapor from the first two effects, little evaporation is left to be accomplished in the last two effects. This has gradually developed into the pressure evaporator which is now popular in Czechoslovakia[3] Here a triple effect is operated entirely above atmospheric pressure. The power units are run on rather high back-pressures (25 to 40 pounds gage) and the exhaust used in the first effect, or live steam is used in the first and exhaust in the second. Vapors are taken from the second effect for the vacuum pans, and all vapors from the third and some from the second are used in the various heaters about the plant. With this decrease in number of effects and disappearance of condensers and vacuum pumps, operation is greatly simplified; higher vapor temperatures make possible complete utilization of vapors in heaters; higher boiling temperatures result in lower viscosities and higher coefficients; and the net result is a very considerable saving of steam, lower first costs, and lower operating costs. This system does not use much, if any, less steam than a quadruple effect evaporator with two pre-evaporators and proper use of vapors, but it is cheaper to install and easier to operate. No damage to the color of the juice results from the higher thick-juice temperatures. The universal practice in this country of boiling with live steam in the vacuum pans means that in American mills the heating surfaces now installed are too small to permit boiling in the pans with vapors. Hence this system of pressure evaporation has made little progress in this country.

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[2] This term has a variety of meanings In the sugar industry (beet or cane) it means only the apparatus for boiling to grain. In some industries, it means any single effect evaporator, for any purpose. In the salt industry, it means any evaporator, single or multiple. In this book, the term is used only as used in the sugar industry.

[3] Kerr, Bull. No. 149, La. Agr. Exp Sta, 59 (1915); Koppl, *Z Zuckerind cechoslov Rep*, *47*, 395-8 (1923); Linsbauer, *ibid*, 574-80, Heinze, *Apparatebau*,

Cane Sugar.

The use of the residual bagasse as fuel, the large amount of power used for crushing cane, and the simpler process of cane sugar manufacture as compared to beet sugar, have resulted in a much simpler steam flow-sheet in the average cane mill. The literature on evaporation and steam economy in the cane mill is not as voluminous as for beet mills, but is too extensive to reproduce here.[4] There is little demand for steam for heating except for heating the feed to the evaporator itself. Straight triple, quadruple, or quintuple effects (the first now used only in older mills, the third a relatively new development) of the standard vertical evaporator, using exhaust or live steam to heat the feed, are the rule. Vacuum pans are usually heated with live steam. In general, there is much room for improvement in this field.

SALT MANUFACTURE.

Most of the other industries using evaporators have lagged far behind the beet sugar industry. A fair comparison is the salt industry. While the first vacuum evaporator for sugar was built in 1812, and the first multiple effect in 1843, the first single effect vacuum evapo-

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35, 177-9, 188-9 (1923); Anon., *Centr. Zuckerind.*, *32*, 142 (1923); Ulrich, *et al.*, *ibid.*, *32*, 218-22 (1924); Tschaskalik, *ibid.*, 307-10; Black, *ibid.*, 1130; Willcox, *Facts about Sugar*, *18*, 522-3 (1924).

[4] See chapters in all the handbooks on cane sugar manufacture and numerous journal articles, but especially Webre, "The Heat Balance in the Cane Sugar Factory," U. S. Cast Iron Pipe and Foundry Co., 1924.

rator for salt in the United States was built in 1885, and the first multiple effect a little after 1890 The writer does not have data available for the first evaporators in the salt industry abroad, but they were not much earlier than in the United States.

In some districts, natural brines are used for the manufacture of salt. These may or may not be fully saturated In a few plants (notably near Salt Lake City and on San Francisco Bay) crude salt is made by solar evaporation. When high-grade table or dairy salt is desired, this crude salt is re-dissolved to a saturated solution. By far the larger number of salt plants operate on an artificial brine made by running fresh water into a cavity in a bed of rock salt and pumping up a saturated brine. Thus salt evaporators usually have as feed a cold, saturated solution of sodium chloride, often contaminated with consideration calcium sulfate, and always carrying some chlorides of calcium and magnesium.

It is not commercially feasible to treat such brines to remove calcium sulfate, hence the vertical tube evaporator is standard The central downtake has been used to a large extent, but a recent installation has been made in which the annular downtake or basket type has been very successful. Singles, doubles, and triples are found, but in recent years a few quadruple effects have been installed.

The average salt plant uses considerably more steam for evaporation than can be produced as exhaust. All the power needed for the plant is usually generated in non-condensing prime movers in which not over 10-15 pounds back pressure is carried. The make-up steam is rarely generated in a separate battery of low-pressure boilers; usually it is high-pressure steam stepped down in a reducing valve. The back pressure carried on the power units limits the temperature of the first effect. By the time four effects are installed with four losses of temperature drop due to elevation in boiling point, the available temperature drop per effect remaining for evaporation is very small, circulation is poor, coefficients low, and scaling rapid. Propellers in the downtake to aid circulation are common in all types of salt evaporators, but are necessary in quadruple effects.

"Salting Up."

As a salt evaporator operates, salt crystals begin to grow on the heating surfaces as well as in the liquid. They usually begin to form immediately below the top of the tubes. This growth of crystals increases rapidly, extending down the tube as well as out from the tube wall, till the tubes are completely closed. Hence salt evaporators must be shut down and boiled out with fresh water at intervals of 12 to 24 hours to remove this deposit of salt.

The cause of this is obscure, but it is not the same as scaling. On pages 198-99 it was shown that for typical scaling processes, $\frac{1}{U^2}$ was a straight line function of time. Figure 107 shows one experimental run in the author's laboratory, using saturated sodium chloride solution in a vertical tube basket type evaporator. Curve 1 is the coefficient as determined, Curve 2 is $\frac{1}{U^2}$ plotted against time. This evidently does not obey equation (86) and hence is not due to the phenomena discussed under the head of scale formation.

Whatever the cause of "salting up," it seems fairly certain that it can be minimized, if not actually prevented, by sufficiently vigorous circulation. This statement is based on general experience, and also on a number of unpublished experiments like Figure 1067 from the author's laboratory. We find, for instance, that single effects with a good temperature drop but without propellers salt up more slowly than triple effects with moderate circulation. Increasing propeller speed usually decreases salting. Tests in the writer's laboratory, in which the same evaporator was operated under identical conditions with and without stirring, showed a cycle more than twice as long when a propeller was used as when operating without it.

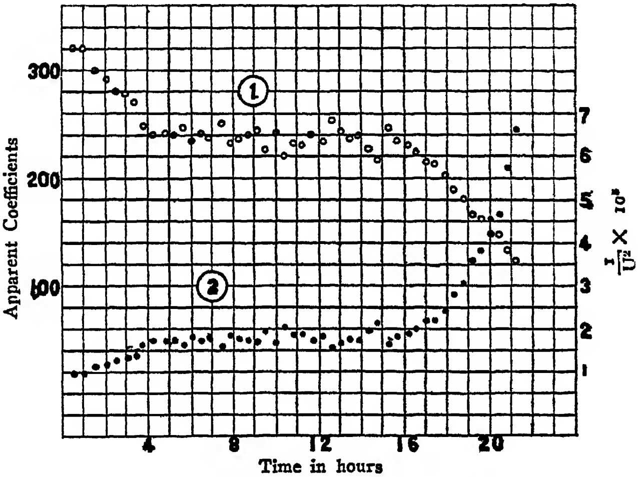

FIG. 107.—Performance of salt evaporator.

Many salt brines, especially those of Michigan, contain considerable calcium sulfate. This comes out mainly in the first effect, where the temperature is the highest and the solubility of calcium sulfate lowest. The deposition of this scale aggravates the conditions described above, as the scale serves as a cement to bind the salt grains to the tube. In such cases, considerable relief is obtained by boiling with water, but sulfate scale must be drilled from the tubes at intervals—in Michigan at least once a week and sometimes twice a week.

Improved Practice.

It would seem that several ways were open for improving practice in salt evaporation. The low price of salt and the small margin of profit is putting considerable pressure on manufacturers to increase the number of effects, so that three effects are now standard and four effects are being considered in new installations. Further progress along this line cannot be made unless the total temperature drop is increased by increasing the steam pressure to the first effect At present most plants feel that evaporator bodies must be of cast iron to prevent discoloration of the salt. The very large bodies now popular (15-23 feet in diameter) and the use of a central downtake, quite effectively prevent the use of higher pressures in the first effect steam space, due to the inherent difficulties of design where cast iron is used. The basket type may be designed for considerably higher pressures in the first effect basket, as this may be made of copper or other metals much more reliable than cast iron. This may make possible one more effect in some cases, although the problem of back pressure on the power units may present complications. A plant whose power units are already installed may not be able to raise the back pressure to 30-40 pounds; but the introduction of this exhaust in the second effect may be provided for in the design. Where a new plant is planned, such high back pressures should be seriously considered

The above considerations are only a partial solution of the problem, as they still call for throttling of the high-pressure make-up steam. Considerable power may be available from expanding this steam from boiler pressures to evaporator pressures. This power may be applied in two ways—by operating a thermocompressor and thus increasing economy, or by causing artificial circulation and thus increasing capacity.

Thermocompression is the least attractive method. Even in a quadruple effect the temperature drop across the first effect is too high for a nozzle to be of value, and the loss due to elevation in boiling point makes a turbo-blower unattractive. Coupled with the high first cost of a turbo-blower installation, this practically rules out thermocompression except in a very special case.

Artificial circulation is the more attractive proposal. The make-up steam should be expanded through a turbine, generating power to drive propellers or circulating pumps. Circulating pumps have a higher first cost, but because of their greater efficiency they will usually pay in the long run Propellers should be driven at as high speeds as possible—the more power absorbed in the propellers, the more circulation there is, the better the rate of heat transfer, and the longer the cycles between boiling out. Too many factors still unknown are involved to make possible a calculation of how far it is profitable to spend money for equipment for circulation, but most salt evaporators could be made to absorb much more power than is customary. A single central turbo-generator and motor-driven circulating devices make an arrangement preferable to turbine-driven circulating units; as either circulating pumps or propellers run altogether too slowly to be driven by a turbine.

Feed is almost always in parallel (i.e., fresh brine fed to each effect) and the feed is almost never preheated. Considering the importance of steam cost in salt manufacture, it would seem almost a foregone conclusion that feed should be heated as far as possible in vapor heaters. In the few cases where an unsaturated brine is used, the brine is also usually impure and a considerable proportion has to be discarded as "bitterns." These bitterns are often further evaporated for calcium and magnesium chlorides, hence they may be discharged from the evaporator as hot as desired without loss of heat. In these cases backward feed is indicated and should show a considerable saving.

ELECTROLYTIC CAUSTIC.

This problem is similar to salt manufacture; simpler, in that there is no calcium sulfate and no tendency to salt up; more difficult in that the final product is very viscous and has a large elevation in boiling point.

Operating Conditions.

Electrolytic cells are usually fed with a saturated brine which has been chemically treated to remove calcium and magnesium salts. It is electrolyzed to such an extent that the effluent from the cells contains 10-12 per cent NaOH, 15-13 per cent NaCl; and 1 per cent or less Na_2SO_4. This cell liquor is usually at 120-140° F., but is often stored long enough to cool to room temperatures. It is evaporated to roughly 50 per cent solids in evaporators and then sent to direct-fired pots for finishing to fused caustic. At 50 per cent solids only about 1 per cent NaCl and Na_2SO_4 are left in solution, most of the sodium chloride and a little of the sodium sulfate being thrown out on concentrating. Some plants run the cell liquor through refrigerating apparatus to freeze out sodium sulfate as Glauber's salt. The salt thrown out of solution in the evaporators is usually re-dissolved to form cell liquor. The elevation in boiling point of 50 per cent caustic solutions is about 90° F. at atmospheric pressure and 80° F. at 26 inches vacuum. There is a considerable increase in viscosity as the concentration increases.[5]

Current Practice.

Two factors limit the number of effects which may be operated on electrolytic caustic liquors—the large elevation in boiling point, and the properties of materials of construction.

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[5] The best collection of data on the important properties of caustic cell liquors is given by Hooker, *Trans. Am. Inst. Chem. Eng.*, *13* (I), 60-76 (1920); *Chem. Met. Eng.*, *23*, 961-8 (1920).

Plate steel, especially where strained, gradually becomes brittle under the effect of caustic soda.[6] This is apparently due to a slight action of the caustic on the iron, forming sodium ferrite and liberating gaseous hydrogen. This hydrogen seems to be absorbed by the amorphous material between the crystalline grains of the metal, thickening the grain boundaries and weakening the bond between the grains. The action is aggravated as the pressure (or temperature) increases

The consequence of this weakness of plate steel is that the pressure of the steam to the first effect must be limited to 30-50 pounds, many operators not caring to go above the lower figure. This so limits the available temperature drop that caustic liquors cannot be finished to 50 per cent in a double effect, and can be concentrated to 37-40 per cent solids only in a double Since the total available temperature drop between 30 pounds steam and 26 inches vacuum is only about 150° F., the reasons for these statements are obvious

Usually, therefore, this material is concentrated in a double effect, equipped with such extra vapor piping as to permit operating as two single effects when desired. Cell liquor is fed to the evaporator in forward feed, and concentration proceeds till the second effect reaches 35-40 per cent solids. The evaporator is then operated as two singles till the second effect reaches 42-50 per cent solids, when the batch is withdrawn. In some cases the batch is carried to 50 per cent solids directly in the second effect, operating as above; in other cases, the second effect is operated as a single while concentrating from about 37-38 per cent to 42 per cent, the batch withdrawn at 42 per cent solids, and sent to a special single effect known as a "high concentrator" (Figure 108) for finishing to 50 per cent solids. Here cast iron tubes are used to permit high steam pressures. To reduce the difficulties of securing cast iron tubes to tube sheets, the tubes are closed at one end ("porcupine" or field tubes) Inside each cast iron tube is a smaller tube, secured to a separate tube sheet, to remove air.

Salt may be removed by salt filters or by continuously operating settlers. In some cases, all the feed to the second effect is pumped to storage and settling tanks, and drawn from these storage tanks to the second effect. The same procedure is sometimes used between the second effect and the high concentrator. Due to the increasing viscosity of the liquor as concentration increases, the salt crystals become finer and their settling time longer; hence the final product of the evaporators must be withdrawn to storage tanks for several hours' settling before going to the pots.

One curious phenomenon is that caustic evaporators seldom if ever "salt up." The presence of the sodium hydroxide, or the viscosity of the medium, seem to effectually prevent this action which is so troublesome in ordinary salt evaporators. Consequently, although vertical tube evaporators are usually preferred because of their more vigorous circulation, horizontal tube evaporators have proved entirely satisfactory for concentrating electrolytic caustic. The importance of the viscosity of the medium in relation to "salting up" is shown by the fact that glycerine evaporators seldom salt up, although they too precipitate considerable salt during concentration. Since there is no calcium sulfate in caustic cell liquors, there is no trouble from scale and the vertical tube type is not necessary on this score.

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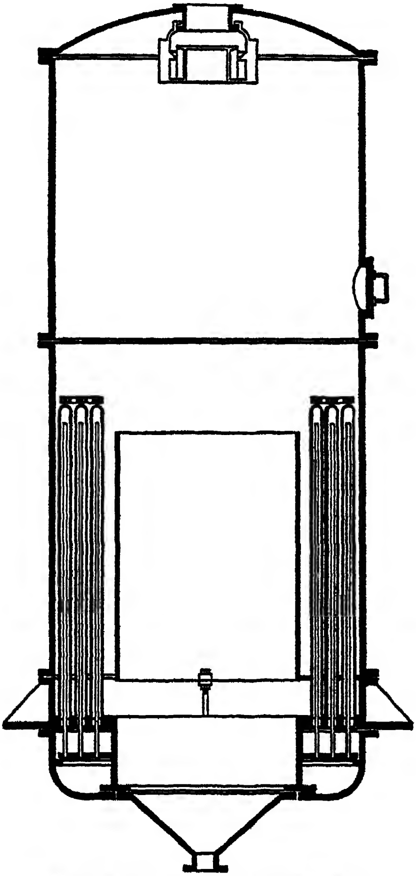
[6] Andrew, *Trans Faraday Soc.*, *9*, 316-29 (1914).



Courtesy Buffalo Foundry and Machine Co.

Fig. 108.—High concentration evaporator for caustic.

Possible Improvements.

The first and most obvious possibility for improvement is backward feed while running as a double. The feed is usually so cold that a large proportion of the steam used (20 per cent in one case that came to the writer's notice) is used for heating feed. Counter-current vapor heaters are not attractive, as their design offers many difficulties. The practice of using storage tanks between the first and second effects, thus effectually cooling the feed to the second effect, is also an error. Backward feed would have an additional advantage in that during double effect operation the highest concentration would be in the first effect, thus considerably lowering viscosity and thereby improving heat transfer coefficients.

At present, no metal is available which will permit safe operation at higher steam pressures, so no increase in number of effects may be expected on this basis If forced circulation devices could so increase the coefficients that the effects could operate on a smaller temperature drop, triple effects might be possible, and would save more steam than the circulating devices would consume.

In those electrolytic plants where the current for the cells is generated by steam, there is ample exhaust for evaporation in double effect. A metal which would permit more effects by permitting higher pressures would be of no advantage, for back pressures higher than 30-50 pounds would not be practical.

In those plants whose current is obtained from hydro-electric stations, the condition is quite different Here steam must be raised especially for evaporation; and such steam is usually the major part of all steam used in the plant. Therefore, any method of reducing its quantity represents a saving of considerable magnitude.

If it were not for the difficulty with metal, the obvious method would be to increase the pressure of the steam to the first effect and add more effects. This is not now possible, though it has been suggested that nickel steels in contact with caustic do not show the brittleness which is common with ordinary plate steel. The other alternative is to expand the steam through a power unit instead of through a throttle valve, and use this power in the evaporators to cause circulation. Even more than in the case of salt evaporators, thermocompression is not attractive because of the extreme elevation in boiling point of caustic liquors. It is true that thermocompression evaporators have been successfully operated on caustic in Europe since the war, but such comparisons mean nothing because of the great discrepancy between the cost of fuel in Europe and in this country, especially as compared to overhead expense.

DISTILLED WATER.

Distilled water evaporators in general do not call for any special features of design or operation, except such provisions for removing scale as may be called for by the hardness of the feed water. Usually as many effects as possible are used to reduce the cost of the resulting distillate.

Power Plant Make-up Evaporators.

This particular phase of the distilled water problem has come into prominence in recent years with the increasing tendency to operate central station boilers on distilled water only. The method has considerable advantages and is applicable to any power plant using surface condensers.

Unless appreciable amounts of process steam are used, it is possible to so arrange the power plant that 90 to 98 per cent of all the steam raised in the boiler is available as distilled water for boiler feed. If this 2 to 10 per cent of make-up is also distilled water, the boilers never have to be scaled or blown down, their life is longer, and their operation much less expensive.

There are two principal systems used for operating these make-up evaporators; known as the high-heat-level and low-heat-level methods, respectively. The second is less common but simpler. The evaporator operates under a vacuum, the steam to the first effect is exhaust from non-condensing auxiliaries or is steam bled from the appropriate stage of the main turbines. Since not only the steam used but the vapor formed are available as distilled water (assuming a surface condenser on the evaporator), a single effect produces theoretically two pounds of water per pound of steam; a double, three pounds, etc. This is to be reduced in practice by steam necessary to heat evaporator feed, to make up for radiation, etc. All the heat leaving the last effect of the evaporator in the vapor must be thrown away in the discharge from the evaporator condenser. This method has the advantage of working at low boiling points where the solubility curve of calcium sulfate, if not concave upward, is at least so flat that scale troubles are minimized.

The high-heat-level system operates usually with a single effect at atmospheric or higher pressure, heated with direct boiler steam. If very large amounts of make-up are to be furnished (10 to 20 per cent) two or more effects may be used. One effect will furnish satisfactory economy for the average power plant. The vapors from the evaporator are condensed by the cool distilled water from the hot-well of the main power condenser, thus returning to the system as heat in the feed all the heat supplied to the evaporator, except that lost by radiation and discarded in the evaporator blow-off. This method demands such a temperature difference between the evaporator vapors and the condensate from the main power units, and such a ratio between this condensate and the make-up, that the condensate will suffice to condense the evaporator vapors. Hence with evaporator vapors at 215-220°, main condensate at 90-100°, and 5 per cent make-up (20 pounds condensate per 2.5 pounds vapor) the condensate is obviously ample and one effect will suffice. If 20 per cent make-up is demanded, or if the main condensate is very warm, more than one effect may have to be installed to cut down the vapors from the last effect to such a quantity as will be condensed by the product of the main power unit condensers. This is a peculiar case, in that the number of effects is dictated, not by the steam economy needed, but by condenser conditions.

Since all the scale which would otherwise be formed in the boiler is now concentrated on the relatively small heating surface of the evaporators, and since these evaporators operate in the temperature range where calcium sulfate causes serious scale troubles, power plant evaporators must be selected with scale removal as the principal controlling condition. All types have been used, but the most popular at present seems to be those where scale is cracked off by turning cold water into the heating coils, such as the Griscom-Russell (Figure 86), and similar types described on page 202. It should be noted that such types operate satisfactorily only on hard and brittle scales; soft scales cannot be removed by such methods. If the raw water used to feed the evaporators has its hardness mainly as calcium sulfate, the scale will be brittle and can be cracked off. If the water contains little calcium sulfate, and much calcium carbonate and suspended organic matter, these evaporators will be unsatisfactory. Such an evaporator as the standard vertical or Yaryan, which can be cleaned with a standard tube cleaner, is indicated in such cases.

PAPER PULP WASTES.

Of the various processes of producing paper pulp, only one (the ground wood process) produces no waste liquors. The soda, sulfite, and sulfate processes all produce wastes which are recovered either because of valuable constituents or because, if not recovered, they may be public nuisances.

Soda Pulp.

In this process, wood chips are cooked with a caustic soda solution of about 12° Bé After cooking, the digesters are blown down and the pulp washed In the poorest practice, the pulp is blown into a tank with a screen bottom, and the pulp washed clean with a hose. This will give a liquor to be evaporated of a density of 4° Bé. Better practice involves the use of indirect heat in the digesters and an Oliver filter or similar device for washing the pulp, and results in a waste liquor of 9-10° Bé.

These liquors, usually called "black liquor," contain all the resin and lignin of the wood, partly decomposed, and all the alkali used in the cook. They are evaporated because of the value of this alkali. Because of the variable nature of the solids present, the relation between total solid content and specific gravity of the liquors is variable, and they are usually controlled by the spindle alone.

The soda process is oftenest used on non-resinous woods, and in such cases there is little or no tendency to foam. There are no scale-forming constituents, little elevation in boiling point, and the finished material is not excessively viscous. The liquors are, therefore, usually evaporated in standard horizontal tube evaporators. Quadruples are common, with straight forward feed. The steam is largely boiler steam, often raised in waste-heat boilers heated by the incinerator gases. The power demands of a pulp mill are relatively small and variable, and many pulp mills have hydro-electric power or direct water-power. Consequently exhaust steam available for evaporation is not only small in amount, but variable.

The liquor is concentrated to 33-34° Bé. (about 40 per cent solids), at which density it is fed to rotary incinerators. These are fired with additional fuel, though the organic matter in the liquor furnishes some heat. The resulting "black ash" is leached and the solution recausticized for use in the digesters. The hot gases from the incinerators go to waste-heat boilers mentioned above.

The horizontal evaporators used have no special characteristics. Cast iron bodies are usual, and the heating surface is often standard iron pipe. No catchalls or special fittings are needed. Because of the density and viscosity of the finished material, the thick liquor is usually finished in the first effect. Many mills feed in the order II-III-IV-I, a few feed IV-III-II-I. In other words, the methods indicated by theory as desirable are usually followed.

If the wood used is resinous, the black liquor tends to foam. In this case, horizontal evaporators must be built with high bodies and deep banks of tubes, and the evaporator must be run with a low liquor level. This greatly decreases capacity, as the upper tubes merely break the foam and do little real evaporation. The Yaryan evaporator has been widely and successfully used on foaming black liquors, its high velocity effectually preventing foam.

Sulfate Pulp.

In this process, the cooking liquid is a mixture of sodium hydroxide, sodium carbonate, and sodium sulfide. The sulfide is renewed by adding sodium sulfate and reducing it with carbon—hence the name "sulfate" process. It is usually used on resinous woods. The content of inorganic material in the liquor is higher, and hence there are more difficulties in evaporation.

One problem in evaporator operation in this process is variable feed density due to inefficient washing. The difference between an 8 per cent solid content and a 6 per cent solid content is that the latter carries 50 per cent more water which must be evaporated. Sulfate pulp is usually washed in a diffusion battery of 4-8 cells, and good practice gives a uniform liquor of high concentration. According to the care in washing, the thin liquor may be from 8 to 16° Bé., and more or less warm. If hot condensate from the evaporators is used for washing pulp, a considerable saving of steam may be effected through high feed temperatures. Some mills have feed as hot as 130° F.

The liquor is much more viscous than soda-pulp black liquor of the same density, has more tendency to foam, and has a higher elevation in boiling point due to its larger content of inorganic solids. There is more non-condensable volatile material present, so that venting is difficult. There is little tendency to scale. Standard horizontal tube evaporators are usually used, though there are some verticals and some Yaryans used The high viscosity of this material is a handicap to the Yaryan. Because of the elevation in boiling point, few quadruple effects are used, the triple being the commoner form. These are all usually fed forward, in spite of low feed temperatures and high viscosity at the finish.

Evaporation is not carried so high as with soda-pulp liquors—24 to 28° Bé being the usual limit At 28° Bé. sulfate pulp is as viscous as soda pulp liquor at 33° Bé Usually at this density, the liquor goes to a "disc evaporator" This is a horizontal cylindrical sheet iron vessel with a horizontal shaft through the axis. The evaporator is half-filled with thick liquor, and hot gases from the incinerator pass through the upper half On the shaft are sheet iron discs, which rotate with the shaft and carry the liquor up into the hot gases. In this evaporator, the liquor is carried to about 36° Bé., and from this it passes to the incinerator. From the incinerator it is discharged, hardly an ash, more like a viscous pasty solid. It is mixed with the sodium sulfate which must be added to make up losses and charged into a smelting chamber where it is fused, the sodium sulfate reduced to sodium sulfide, and some sodium carbonate formed. This fused melt is run into water, dissolved, recausticized, filtered, and re-used.

The blast in the smelting furnace (which also heats the incinerator) is so strong and carries so much dust that if waste-heat boilers are used after the incinerator as in soda-pulp mills, they must be cleaned daily. Hence, the disc evaporator replaces the waste-heat boilers, and the evaporators proper must be run on direct boiler steam. The liquor is so viscous and pump up-keep so troublesome that few mills even try to finish in the first effect, to say nothing of feeding all the effects backward.

It would seem that this was an ideal case for forced circulation—boiler steam used almost exclusively, cast iron bodies which cannot take high initial pressures, and a viscous material. Because of conditions in this department of a mill, it may be that positive circulation by pumps is impractical, but propellers in a vertical tube evaporator would seem desirable. Backward feed is strongly indicated, but as discussed above, is probably not practical.

Moore[7] has described a method of evaporating and fusing sulfate liquor, in which the liquor is concentrated to 50 per cent solids, mixed with the make-up sodium sulfate, superheated, and sprayed into a furnace maintained at a high temperature. The flash due to releasing the superheat explodes the drops, and radiation from the furnace burns them and fuses the residue in one operation. This process is used in the mill of the Brown Company at LaTuque, Quebec

[7] *Trans Am. Inst. Chem. Eng*, *12* (II), 101-38 (1919).

Sulfite Pulp.

The active cooking reagent in this process is calcium-magnesium bisulfite. The active material cannot be recovered, few useful products can be made from the liquor, the majority of sulfite mills are in isolated districts, and the greater proportion of all sulfite liquor produced is wasted. A little is concentrated and disposed of as core binder, road binder, or tanning liquor; and in a few cases, mills are so situated that they must evaporate their liquors and burn them to avoid stream pollution. This latter case will probably arise with greater frequency in the near future and it is to be expected that more and more sulfite mills will have to evaporate their wastes. In Europe, much sulfite liquor is fermented for alcohol, but the process has not gotten a foothold in this country.

Sulfite liquor is very difficult to evaporate. The sulfites oxidize to sulfate, giving scale; the liquor gives off sulfur dioxide and therefore there is corrosion both by the liquor and vapors; and the liquor is viscous and has a high elevation in boiling point. In the Robeson process[8] the liquor is first neutralized with lime and then evaporated. In this case the thin liquor is about 4° Bé., and is carried to 31° Bé. Neutralization removes corrosion troubles, but does not affect the rest of the undesirable properties. In this case, the evaporator is usually a standard horizontal evaporator with iron pipe heating surface. A bottom containing some space for mud must be provided. The scale is brittle and can be cracked off ordinary horizontal tubes by heat shock, such as turning a high-pressure stream of cold water on the tubes while they are hot.

Sulfite liquor may be evaporated without neutralizing, in which case there is considerable corrosion and a high SO_2 content in the vapors, especially from the first effect. The thin liquors will be about 10 per cent solids and will be carried to 31-32° Bé. (55 per cent solids). The evaporators may be standard horizontals with extra heavy plates, bronze tube sheets and studs, copper tubes, and bronze fittings. Scale troubles are still severe and a half-round bottom should be used to provide a reservoir for scale and mud. Non-condensed gases from the second-effect steam chest should never be vented into the second-effect vapor space, but sent direct to the condenser. Backward feed would seem desirable, but is almost never used. Finishing in the first effect, at least, would be an advantage, but the greatest improvement would be to feed the thin liquor to the fourth effect, irrespective of how the other effects might be fed. This would liberate most of the sulfur dioxide to go direct to the condenser, and save much corrosion of steam-chest castings and tube sheets.

In some cases, acid sulfite liquor is evaporated in tile-lined machines. These are standard horizontal evaporators with sufficient space left for lining with a course of glazed acid-proof brick.[9] This is not only expensive, but if the brick are not carefully laid, a leak through the tile lining may result, and conditions are then as bad as if no lining were used.

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[8] U. S. Pat. 833,634, 1906.

[9] Zaremba and Mantius, U. S. Pat. 1,106,532, 1914.

SOAP LYES.

There are two types of crude glycerine liquors commonly met—Twitchell lyes and soap lyes. The first is the result of hydrolyzing fats by Twitchell's reagent, in which process the fat is split into fatty acid and glycerine. This gives a concentrated lye (10-11 per cent glycerine), free from salt and rather pure, although the purity varies with the type of fat used and the care exercised in the process. Soap lye comes from the manufacture of soap by saponifying fats with caustic soda and salting out the soap. It is weaker than Twitchell lye (4-10 per cent glycerine), contains considerable salt, and on the average is less pure.

Soap-lye Evaporation.

The results obtained in evaporation depend almost entirely on the care with which the lye is purified. The simplest purification process used is to add lime till nearly neutral, settle, and decant to storage tanks. This results in a very impure lye, which will foam and form scale on evaporation, and gives a low quality crude. Possibly the best practice is represented by the following: Add lime till just alkaline, filter, make barely acid with hydrochloric acid, add alum or copperas as a coagulant, filter, neutralize with caustic soda and add sodium carbonate to precipitate calcium salts, filter, store, make barely acid before evaporating. All types of purification processes may be found between these extremes. The lye going to the evaporator will contain 4 to 10 per cent glycerine, about twice as much salt as glycerine, a little sodium sulfate, and it may be neutral, acid, or alkaline. Acid lyes do not foam so badly, and in acid solution some of the organic impurities steam-distil off, so that a purer crude results. In small soap plants, this will be concentrated to a crude (about 80 per cent glycerine) in a single-effect evaporator. The next larger plants will concentrate in a double-effect to semi-crude (35-40 per cent glycerine) and to crude in a single-effect. The largest plants will concentrate in a double or triple to the point where salt begins to appear; then to semi-crude in a single-effect, and to crude in another single.

As glycerine lyes are concentrated, the solubility of sodium chloride in them decreases and the elevation of boiling point increases. The data for the solubility of salt in glycerine of varying concentrations, and the boiling points of glycerine-water mixtures both pure and saturated with salt, have been determined.[10] The increase in density, viscosity, and elevation in boiling point makes it necessary to concentrate from semi-crude to crude in single-effect in batches.

[10] Carr, Townsend, and Badger, *Ind. Eng Chem*, *17*, 643-7 (1925); von Mayer-Bugstrom, *Z. deut. Oel-Fett-Ind.*, *44*, 417-8 (1924); *C. A.*, *18*, 3501.

About half the sodium chloride is thrown out in concentrating to semi-crude, and half in boiling from semi-crude to crude. This may be removed by salt filters, or by using a half-round bottom in which the salt collects during boiling. At the end of a batch, this salt is flushed out to open filter-bottom receivers. Continuous removal of salt by pumping to salt settlers is not used in this industry, but would seem to offer considerable possibilities.

The evaporators used are generally horizontals, though a few verticals are used. The horizontals are all cast iron with copper tubes. Plate steel must be avoided. The vertical would seem to be the preferable type, but it is probably in disfavor because of the general use of steel tube sheets and downtakes. Foam may occur; and due to the value of glycerine, losses by entrainment must be carefully watched. External catchalls of large size are used, two or even three in series for each body. Salting on the tubes may or may not occur. It is customary to start up evaporators in which salt is to precipitate, by feeding dilute lye at the start, even though the main charge may be semi-crude. Foaming and salting troubles are often connected with imperfect treatment. If sodium sulfate be present (from the use of sulfuric acid instead of hydrochloric in treatments) there will be a decided tendency to scale as discussed on p. 198, and salting will be much more serious. .

It would seem that a vertical tube evaporator of the basket type with forced circulation is indicated. The central downtake type tends too much toward entrainment. The deflector on the basket type is a very efficient entrainment separator. Viscosity, high density, salt in suspension, and possible sodium sulfate scale, all call for high rates of circulation, preferably by artificial means.

Twitchell Lyes.

These lyes are much easier to evaporate than soap lyes, due to their greater purity and freedom from salt. They can be concentrated directly to crude in a double-effect, and the crude is denser than with soap lyes (88-90 per cent glycerine). There is less foam than with soap lyes, though this depends on the care taken in purification. A sludge of sodium sulfate separates toward the end of the concentration. Extra space should be provided in the bottom of the evaporator to accommodate this.

PACKING HOUSE PRODUCTS.

The meat packing industry uses evaporators on many products. Tank water, glue, and gelatin are the principal materials. Small evaporators are also used for beef extract, gland extracts, etc., while many packers also make soap and therefore use glycerine evaporators.

Tankwater.

Packing house refuse is cooked with steam under pressure, and the liquid grease and water drawn off. This is allowed to stand till the grease has separated. The aqueous layer is then sent to heated storage tanks, where it is kept at about 170° F. This step is most important, as during settling last traces of grease rise to the top, and more or less finely divided solids settle. If either of these materials passes to the evaporator, there will be trouble from deposits on the tubes or from foam. The feed liquor will be at 3-4° Bé.

The liquor is concentrated to 26-30° Bé and in this form is known as "stick." Horizontal tube evaporators are used, usually in triple-effect. Since most of the substances in solution are colloidal there is little elevation in boiling point, and the increase in viscosity, while considerable, is not serious. The feed is usually hot enough so that forward feed, which is generally used, is correct. The liquor is sometimes slightly acid, hence the machines must be cast iron with copper tubes of a good weight.

Glue.

Glue is made by leaching bones, hide, or other similar material with hot water. The solution coming to the evaporator is of varying density according to number of cooks and the character of the raw material. Likewise the finish density is variable according to the quality of the stock and the characteristics desired in the finished glue Lower grade glues must be finished to higher densities to obtain the same jelly strength.

Since the material is of very high molecular weight, there is almost no elevation in boiling point. There is no trouble from scale, no great viscosity, but a strong tendency to foam—worse in hide glues than bone glues, and worst in fish glues. The evaporator usually used is the standard horizontal of cast iron with copper tubes. Catchalls are not favored, as they form pockets difficult to clean and therefore sources of bacterial decomposition. High vapor spaces are used as the only precaution against foam. Double-effects are commonest, but singles and triples are also used. As with all foamy materials, the Yaryan has been used with good results. There is no special difficulty in any phase of this problem except the troubles from foam.

Gelatin.

Gelatin is merely a high-grade edible glue, made from carefully selected stock with the especial precautions that are obviously indicated in preparing a food product. The thin liquors may be from 0.5 to 6 per cent solids, averaging about 3.5 per cent. The average concentration of the thick liquor is 14 per cent (20 per cent maximum). There is a decided tendency to foam.

Legal requirements prohibit the use of copper; and as iron is impossible because of its tendency to cause discoloration, gelatin evaporators are usually cast aluminum with aluminum tubes. Steam-chest castings may of course be of cast iron, but all pipe and fittings should be aluminum. Brass or bronze valves and brass pumps are permitted. Especial care must be taken to eliminate pockets where material might collect and decompose. Both glue and gelatin evaporators must be very carefully boiled out before a shut-down, and obviously neither caustic soda nor sodium carbonate should be used on aluminum.

Gelatin evaporators are usually single effects. Some doubles are used, but the high boiling point in the first effect makes trouble with the quality of the product. Some operators use double effects successfully by so operating as to maintain a low boiling point in the first effect. Capacity falls off very rapidly as the final concentration increases, due to the viscosity of the thick material. Concentrating from 3 to 10 per cent may be done at nearly twice the rating obtained in concentrating from 8 to 15 per cent.

Chapter 14.

Costs.

It has been said that one of the important differences between engineering and the pure sciences is that the engineer's solution of his problems must be financially profitable, whereas the scientist is under no such handicap. Unfortunately, in this phase as well as in many others, the data available for the solution of the problem are characterized mainly by their insufficiency, so that rigidly accurate solutions are rarely possible. However, approximate estimates are possible in some cases, and this chapter will attempt to lay down a few general principles.

Economy vs. Capacity.

One of the commonest questions asked by engineers who are only superficially acquainted with evaporators is—"What is the efficiency of this machine?" In the first place, the word "efficiency" is so often used loosely, as in this question, that it has little if any precise significance. In the second place, if it is used in its true sense of ratio of work performed to work equivalent of heat input, it has no meaning as applied to evaporators. The efficiency of an evaporator, according to this definition of efficiency, is zero, since an evaporator does no external work.

The writer prefers, instead of "efficiency," the terms "economy" and "capacity," since these latter terms have a fairly definite meaning. Even so, a hard-and-fast definition is not easy to make. The Test Code for Evaporating Apparatus (Appendix II) accepted jointly by the American Society of Mechanical Engineers and the American Institute of Chemical Engineers, defines these terms as follows: *"Efficiency defined.* Probably the most useful, but not the simplest, rating is, *pounds steam used per equivalent pound of water evaporated from and at (a specified temperature)."*[1] *"Capacity.* The capacity of any equipment covered by this code is best expressed by its heat transfer coefficient, or the heat transfer coefficients of as many parts as can be tested separately. . . . If it is desired to rate the capacity of an evaporator by some such factor (as rate of evaporation), this will only be acceptable if stated in terms of standard United States gallons evaporated per hour, from and at (temperature to be specified), with a total temperature drop of (to be specified) degrees."[2]

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[1] Section 39. The writer would much prefer "economy" to "efficiency" in this section.

It should be noted that for a given number of effects and a given method of feeding, *economy* cannot be varied to any appreciable extent. It is not always appreciated that for a given number of effects and a given feed temperature, economy can be varied considerably by methods of feeding. Thus in the problems of Chapter 10, we find the following comparison:

Problem 1, forward feed...	2.42	pounds evaporated per pound steam.
Problem 2, backward feed..	3.06	" " " " "
Problem 3, counter-current vapor heaters with forward feed	3.47	" " " " "

Hence the simple assumption of a flat figure for pounds of water evaporated per pound of steam per effect, for any and all feed temperatures and methods of feeding, is far from correct. But once the feed temperature and method of feeding is established, *any two* evaporators will have the same economy (except for radiation losses) irrespective of their construction.

On the other hand, with number of effects, feed temperature, and method of feeding once established, extreme variations are possible in the heat transfer coefficients. Boiling point, temperature drop, and design of machine affect the coefficients profoundly.

Changes in capacity are reflected in the size and cost of the evaporator needed to do a certain amount of evaporation. This obviously affects such overhead costs as interest, depreciation, repairs, taxes, etc. Since these costs are an appreciable part of the total cost of evaporating a pound of water, they are of major importance. Labor is usually about the same for any type. Since steam, the only important cost remaining, is a function only of method of feeding and number of effects, it follows that wider variations in cost of evaporation are caused by changes in capacity than by changes in economy, once the method of feeding and number of effects are chosen.

It does not follow, of course, that the machine of the highest capacity is the cheapest. If capacity is obtained by such additions or variations as greatly increase the first cost, repairs, or cost of labor, a machine of somewhat lower capacity but much lower cost may be the cheapest in the long run. Hence the decision in each individual case rests upon many factors, but usually is concerned principally with capacity.

Cost of Evaporator Bodies.

Figure 109[3] shows the cost of single-effect evaporators of various types exclusive of accessories. This figure assumes cast iron bodies

[2] Sections 25, 26, 27.

[3] Fuwa, *Chem. Met. Eng.*, *31*, 185-8 (1924).

and iron tubes. It is interesting to note that the standard horizontal is the cheapest of the common types, though the standard vertical approaches it in large sizes. The greater cost of two of the film types is plainly shown.

Cost of Auxiliaries.

Fuwa[4] gives the following data for cost of accessories:

Condensers—8 to 10 per cent of the cost of the evaporator bodies; more for small single effects (up to 20 per cent), less for large multiple effects (down to 5 per cent).

Vacuum pumps—16 to 20 per cent of the cost of the evaporator bodies; as above, less for large evaporators and more for small ones.

Salt receivers—about 10 per cent of the cost of the bodies.

Liquor pumps, traps, etc.—5 per cent of the cost of the bodies.

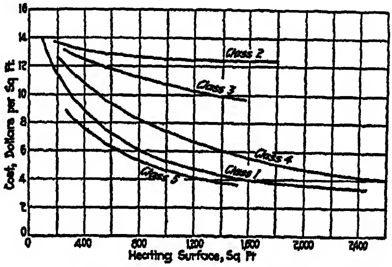

Fig 109.—Cost of evaporator bodies.

Class 1, Horizontal Tube Evaporators; Class 2, Lillie Evaporators; Class 3, Yaryan Evaporators; Class 4, Standard Vertical Tube Evaporators; Class 5, Mantius High Speed Evaporators

All steel construction decreases the cost of the evaporator bodies by about 20 per cent.

Copper tubes instead of iron increase costs by about 5 per cent.

Erection may be anywhere from 15 to 30 per cent.

These estimates will vary more or less with the market prices of labor and metal, and should only be used for rough preliminary estimates.

Cost of Steam.

It should be noted that in some cases it is not correct to estimate the savings due to a change in process by crediting the new arrangement with the total cost of the steam saved. If the boiler plant is already installed and the proposed changes do not reduce the total load by enough to diminish labor or repairs, it is reasonable to credit the new apparatus with the saving in coal only, not with a figure for steam cost which includes overhead and repairs. Since fuel cost is ordinarily somewhere in the neighborhood of half the total steam cost, this point is one of considerable significance.

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[4] *Loc. cit.*

The most difficult problem in this field is the value to be placed on exhaust steam delivered from the power units to the evaporators. The cost of the steam should be divided between the two processes, but an equitable basis is difficult to reach. There are several possible methods,[5] such as dividing the cost of steam between the two on the basis of available power content, available heat content, or some arbitrary division. The second method (heat content) is unfair to the evaporator if total heat is used, as the latent heat is the principal item of value to the evaporator but is not available to the power unit. If this division is made on the basis of heat available for power generation, the basis is fairer. The first method seems the most reasonable. In any case, the evaporator replaces power condensers and vacuum pumps, and hence the evaporator should be credited with the cost of condensing the power unit exhaust.

Operating and Overhead Costs.

These may be taken as the sum of interest, depreciation, repairs, taxes and insurance, and labor.

The item of interest varies with the plant and its method of financing. Some plants, instead of figuring this item as interest, call it "minimum attractive profit," which is really a better name. Therefore a fixed amount cannot be set for this figure.

Depreciation depends on the service, the corrosive action of the liquor handled, and on possible obsolescence. Standard salt or sugar evaporators may last 25 to 30 years without serious repairs other than tube renewals; corrosive liquors may necessitate practically rebuilding a machine in a year or two. For general service in standard lines a life of 15 years is not unreasonable. In the same way, repairs vary enormously with the service; in standard machines on non-corrosive liquors they should not be much over 5 per cent per year. Taxes and insurance should not be over 3 per cent.

Labor is fairly easy to estimate. One man can operate any but the largest and most complicated evaporators. He may have to have helpers if salt receivers are used, or if there are circulation pumps to be attended, but an ordinary evaporator can be run by one man. In some cases, two moderate sized triple effects, without special auxiliaries, are run by one man. Extra labor is needed during scale removal, but this usually comes from the master mechanic's crew.

Economic Number of Effects.

As the number of effects is increased, the overhead costs increase, while steam cost is decreased. Obviously, the total cost will be high with too few effects, due to excessive steam consumption; and high with too many effects, due to excessive overhead. There is a minimum that represents the proper number of effects to use, and this can be

[5] Fuwa, *Chem. Met. Eng.*, *31*, 150 (1924).

readily determined by calculating total cost for two or three arrangements.

It should be noted that any desired number of effects cannot always be installed. Too many effects may use up so much of the temperature drop in boiling point losses that there may not be enough temperature drop left for operation Figure 61 makes possible an approximate estimate of the falling off of coefficients with small temperature drops, and this must be considered in such calculations as the above. The decrease in available temperature drop per effect may necessitate using much lower coefficients (and hence much larger bodies) as the number of effects is increased. This factor alone may be the controlling one, as in the case of electrolytic caustic.

Closely connected with this problem is the question of exhaust pressure where the evaporator is wholly or largely operated on exhaust steam. No rule can be laid down for this except the general rule that such a choice should be made as to give the lowest cost for evaporation and power generation combined Too low back pressure on the power units makes power generation cheap, but decreases the amount of exhaust steam available, calls for more high-pressure make-up steam to the evaporator, limits the number of effects possible, and makes evaporation too expensive Too high back pressure makes evaporation cheap but power generation too expensive. The proper point may only be found by trial and error, knowing all the factors applying to the particular case in hand. For instance, a certain plant uses a relatively large amount of power and has power units whose water rate is very high at even moderate back pressures. The demands in this plant for steam for evaporation are moderate, and the economic balance has been found to be reached at 24 inches vacuum on the power units, the evaporators operating in single effect with steam at 6 inches absolute and vapors at 2 inches absolute. This is an extreme case, and conditions more favorable to the evaporators might be found with other types of power units, but it is the economic solution for that particular plant.

Back pressures much over 30 pounds gage are rare, as most power units use excessive amounts of steam above this range. Data have been published[6] on the performance of small steam engines at high back pressures, when their exhaust was needed for process steam and the engine was used as a reducing valve.

In this connection some recent foreign developments along the line of excessively high boiler pressures are of interest.[7] It has been shown that the design of boilers and engines for the generation and use of steam up to 1500 pounds presents no special difficulties. Plants have been operated commercially under these conditions, and even up to 3300 pounds. The result is that power may be generated so economically from this high-pressure steam that the exhaust may be sent to evaporators or other parts of the process at pressures as high as 150 pounds. This opens up an entirely new field for steam flow sheets of plants needing process steam at pressures higher than can now be carried as exhaust pressures.

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[6] Field, *Chem. Met Eng*, *20*, 18-24 (1919), Flowers, Pepper, and Field, *Chem Met. Eng*, *22*, 318-20 (1922).

[7] Hartmann, *Z Ver deut Ing*, *65*, 663-71, 713-9, 747-53, 848-52, 988-93, 1047-8 (1921); Hartmann, Schmidt, and Wolf, *Z Ver deut Ing*, *66*, 345 (1922); Watzinger, *Papir-Journalen*, *11*, 181-5, 183-200 (1923), *12*, 3-5 (1924); Praetorius, *Chem. Ztg*, *48*, 377-80 (1924), Kruchen, *Centr. Zuckerind.*, *32*, 709-11, 733-5 (1924).

Economical Cycle.

The case of an evaporator which must be cleaned at intervals brings up the question of the proper length of cycle. This may be determined either on the score of maximum production or minimum cost. Assuming that equation (86) holds, the following derivation can be made:[8]

Let X = time of one cycle in hours,
C = time for cleaning in hours,
N = no. cycles per 24 hours $= 24/(X + C)$,
A = total heating surface of evaporator,
L = heat input per pound of water evaporated,
Q = total heat transferred per cycle,
G = pounds evaporated per cycle,
Δt = working temperature drop,
E = pounds evaporated per 24 hours.

Solving equation (86) for U we have

$$U = \frac{1}{\sqrt{aT + b}} \qquad (99)$$

The total heat transmitted per cycle, Q, is given by

$$Q = AU\,\Delta t = A\,\Delta t \int_0^x \frac{dT}{\sqrt{aT + b}} \qquad (100)$$

and for the total evaporation per cycle, G, we have

$$G = \frac{Q}{L} = \frac{A\,\Delta t}{L} \int_0^x \frac{dT}{\sqrt{aT + b}} \qquad (101)$$

If all the constant factors on the right-hand side be combined into one constant $\frac{1}{m}$ we have

$$mG = \int_0^x \frac{dT}{\sqrt{aT + b}} = \left[\frac{2\sqrt{aT + b}}{a}\right]_0^x = \frac{2\sqrt{ax + b} - 2\sqrt{b}}{a} \qquad (102)$$

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[8] Badger and Othmer, *Trans. Am. Inst. Chem. Eng.*, 17 (1), (1925).

Equation (86) may be plotted from experimental data ($1/U^2$ as ordinates against time) and from such a plot the numerical values of a and b may be determined graphically. When these are substituted in equation (102) a definite numerical solution is possible, as the factors included in the constant m are all known.

In order to find the length of cycle which will give the maximum production per 24 hours, we may write

$$E = NG = \frac{24G}{X + C} \tag{103}$$

Substituting the value of G from equation (102) and rearranging,

$$E = \frac{48A\,\Delta t}{aL}\left(\frac{\sqrt{aX + b} - \sqrt{b}}{X + C}\right) \tag{104}$$

In order to find the maximum value of E we may proceed in several ways: (a) we may plot equation (104) and determine its maximum by inspection; (b) we may differentiate equation (104) either directly or graphically and determine from the plot of the derivative where it is equal to zero, or (c) we may differentiate equation (104), set the first derivative equal to zero, and solve for X. Method (c) is very clumsy for this equation, and the quickest and simplest method is (a).

As an illustration we may take some data from the evaporation of hard water. The experiments were run in an experimental Yaryan evaporator having one iron tube, 2½ inches in diameter and 50 feet long, encased in a steam jacket. The water contained 280 p.p.m. of $CaCO_3$ and 156 p.p.m. of $CaSO_4$. The boiling point was 210° F. and the steam temperature 286° F. Curve 1 (Figure 110) shows the data as taken, expressed as coefficient *vs.* time. Curve 3 (Figure 111) shows the plot of equation (86) derived from this data. From this line we find the constants of equation (86) to be

$$a = 0.212 \times 10^{-6}$$

$$b = 2.55 \times 10^{-6}$$

Substituting in (104), assuming $C = 3$ hours and letting p equal the constant factors on the right-hand side of the equation, we have

$$\frac{E}{p} = \left(\frac{\sqrt{0.212X + 2.55} - 1.595}{X + 3}\right) \tag{105}$$

The plot of the right-hand side of this equation is shown in Curve 5 (Figure 112), and the maximum is seen to be at 15 hours. That is, if this machine is cleaned every 15 hours the total production will be a maximum. Curve 6 is a graphic differentiation of Curve 5, and this shows a value of zero (maximum production) at 15 hours. It is interesting to note the very flat portion of Curve 5 near the maximum.

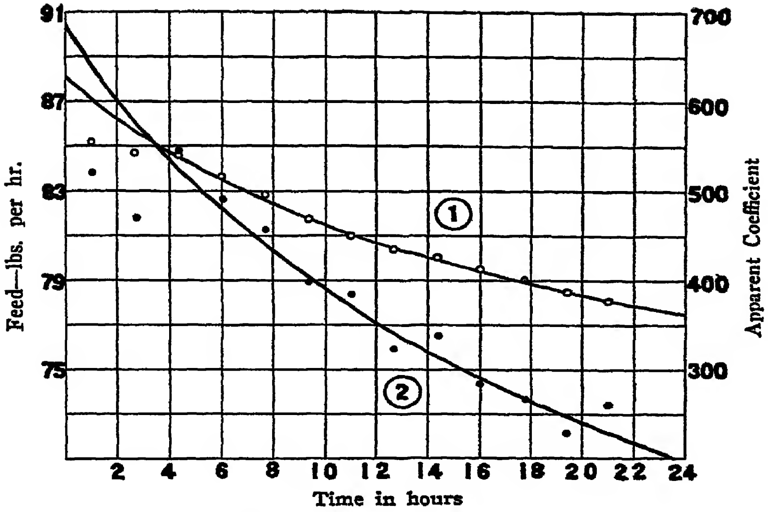

FIG. 110.—Scaling of Yaryan evaporator on hard water.
Curve 1.—Heat transfer coefficients.
Curve 2.—Rate of feed.

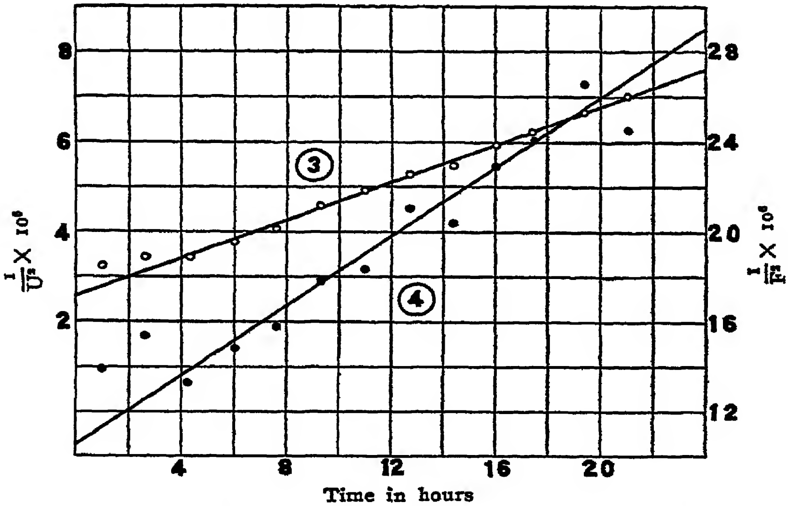

FIG. 111.—Performance of evaporator on hard water.
Curve 3.—$1/U^2$ vs. time.
Curve 4.—$1/F^2$ vs. time.

The total evaporation per 24 hours is not much different for cycles between 10 and 24 hours. Similar curves have been obtained for other cases. In other words, the evaporation per 24 hours would not be greatly different whether the cycle were 12, 16, or 24 hours. Hence this machine could be boiled every 12-hour shift, every two 8-hour shifts, or every 24 hours, without affecting the capacity per 24 hours. This is a function both of the steepness of the $1/U^2$ curve and of the time for cleaning. With steeper $1/U^2$ curves and longer times for cleaning, the maximum is more plainly marked.

In many cases in practice the determination of a heat transfer coefficient is not convenient. If the temperature drop be maintained constant, and if other factors (such as ratio of feed to discharge) remain constant, the coefficient may be taken as proportional to the feed, or to the condensate, or to any other quantity which can be measured easily and which is proportional to the evaporation. A flow-meter on the feed would probably be most convenient.

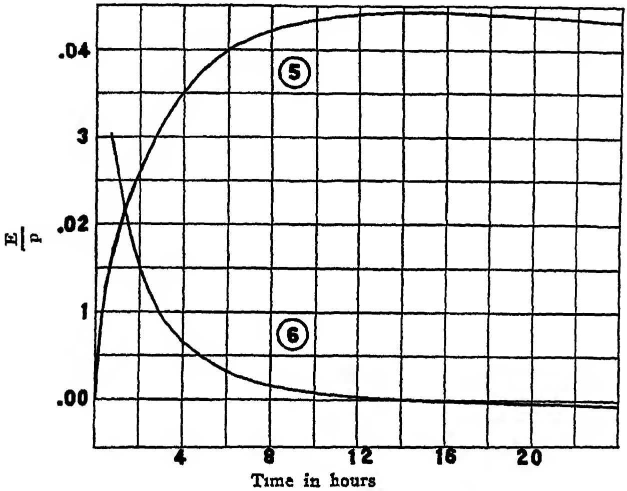

Fig 112.—Economical cycle for evaporator of hard water.

Curve 5—$\frac{E}{p}$ from equation (105).

Curve 6.—First derived curve from curve 5.

Curve 2 (Figure 110) represents rate of feed against time for this same run and Curve 4 (Figure 111) is the $1/F^2$ curve analogous to equation (86). Since Curve 4 is also a straight line, the constants a and b may be derived from it, and substituted in equation (105) as before. This will give the length of cycle for maximum *feed,* but since by assumption evaporation is to be a constant fraction of the feed, this will give the maximum evaporation also. Due to variations in temperature drop and in discharge ratio, the $1/F^2$ curve will not be as accurate as the $1/U^2$ curve, but the accuracy of the integral curve will be about the same in both cases.

The cycle of maximum production may not be the cycle of minimum cost. This depends on the relation of operating to overhead costs, on the ratio of boiling time to cleaning time per cycle, and on the ratio of cost per hour for labor for cleaning to cost per hour for labor for operating. Low overhead costs and high costs for cleaning tend to longer cycles, and vice versa.

If we assume that interest, depreciation, repairs, insurance, and taxes may be represented by a flat sum per day, we may make the following analysis:

Let o = overhead cost in dollars per 24 hours,
d = labor cost during boiling in dollars per hour,
e = labor cost during cleaning in dollars per hour.
f = steam cost per pound of evaporation.

Then the total cost per day is

$$o + dNX + eC + Ef$$

and we may write

$$\text{cost per pound evaporated} = \frac{o + dNX + eC}{E} + f \qquad (106)$$

Substituting $\frac{24}{X + C}$ for N, and the value of E from equation (104), we have

$$\text{cost} = \frac{aL(X + C)(o + eC) + 24adLX}{48A\,\Delta t(\sqrt{aX + b} - \sqrt{b})} + f \qquad (107)$$

This may be evaluated for any given case, plotted, and the length of cycle giving the minimum cost may be determined by inspection. The possible variations in the actual values of the constants in this equation are so great that it has not been thought worth while to work out an illustrative problem.

Heater Formulas.

The method for determining the economical velocity in a heater was given in Chapter 7. Lewis, Ward, and Voss[9] have developed a number of equations for this and similar cases, based on equation (48), and show that

$$V_o = \sqrt[3]{\frac{46.3\,bE}{acf\rho D\left(1 + 50\frac{D}{L}\right)}} \qquad (108)$$

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[9] *Ind. Eng. Chem., 16*, 461-2 (1924).

where V_o = optimum velocity in feet per second.

b = total cost of heater, installed, in dollars per foot of length per tube.

E = per cent annual charges on cost of heater.

a = cost of energy in dollars per horsepower hour delivered to the fluid, including cost of pump maintenance, etc.

c = total hours of operation per year.

f = pipe friction factor.

ρ = density of liquid in pounds per cubic foot.

D = inside tube diameter in feet.

L = length of one tube in feet.

It should be noticed that the optimum velocity is independent of temperature drop, of the heat transfer coefficient (either liquor film or overall) and of viscosity.

If a saturated vapor be available, condensable at a temperature T, in an apparatus having an overall heat transfer coefficient U, and there is a limited amount of cold liquor available, it may not be economical to heat the liquor as near to T as possible. In the above-mentioned paper, it is shown that it is not economical to heat the liquor above a temperature t_2, given by

$$t_2 = T - \frac{M}{UR} \tag{109}$$

where M is the total cost, in dollars per hour per square foot, of operating the heater, and R is the value to the plant of 1 B t.u. as here recovered. If the vapor must all be condensed in any case, the cost of such condensation per B.t.u. extracted must be added to R.

Other similar formulas are developed, the one of most general importance being the one which gives the maximum temperature to which a cold fluid should be heated in a counter-current heat interchanger.

T = temperature of hot fluid.

t = " " cold "

Subscript 1 = temperature of entrance to heat exchanger.

Subscript 2 = " " exit " " "

W_T = pounds per hour of hot fluid.

W_t = " " " " cold "

S_T = specific heat of hot fluid.

S_t = " " " cold "

M, U, and R as above.

Then, the maximum value which t_2 should have is found by solving the following:

$$(T_1 - t_2)\left[\left(\frac{S_T W_T}{S_t W_t}\right) T_1 - \left(\frac{S_T W_T}{S_t W_t} - 1\right) t_1 - t_2\right] = \frac{M S_T W_T}{U R S_t W_t}(T_1 - t_1) \tag{110}$$

Appendix 1.

Flow of Fluids.

This material is all taken from Section V, Chapter 3, of "Principles of Chemical Engineering" by Walker, Lewis, and McAdams (McGraw-Hill).

Fanning's Equation.

The basic equation for the flow of fluids through pipes is

$$\Delta p = \frac{2fL\rho V^2}{gd} \qquad (111)$$

where Δp = loss of head in pounds per *square foot.*
f = friction factor (see below).
L = length of pipe in feet.
ρ = density of fluid in pounds per *cubic foot.*
V = average velocity of fluid in feet per second.
g = 32.2 feet per second per second.
d = actual inside diameter of pipe in *feet.*

This equation holds for gases as well as for liquids.

The value of f in equation (111) is to be taken from Figure 113, in which the values of f are plotted as ordinates against $\frac{DVs}{Z}$ as abscissas. Here

D = actual inside pipe diameter in *inches.*
Z = viscosity of fluid in centipoises.
s = specific gravity of fluid.

The steep straight line to the left of the chart represents non-turbulent or viscous flow. The two curved lines to the right represent turbulent flow; the upper being for steel or cast iron pipes, the lower for brass or copper (probably for lead also). If the pipe is much roughened by corrosion or incrustations, f must be correspondingly increased. It should be noticed that the condition of the pipe walls makes no difference in viscous flow.

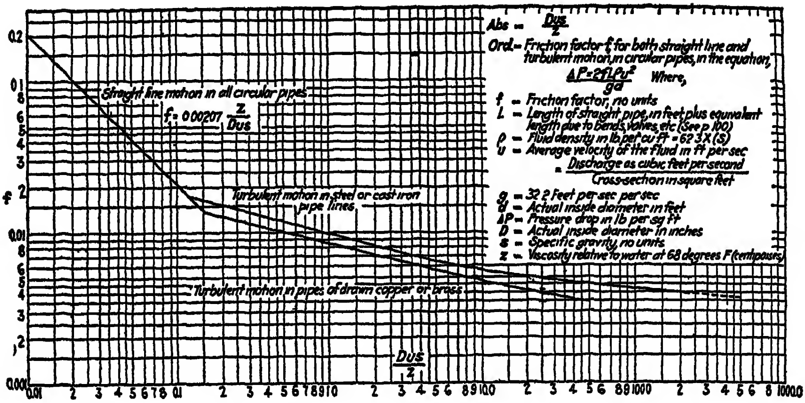

FIG 113.—Friction of liquid in pipes

Other Losses.

Where there is a sudden *increase* in pipe diameter, the loss in head is given by

$$\Delta h = \frac{(V_1 - V_2)^2}{2g} \qquad (112)$$

where Δh = loss in head in *feet of the fluid*
V_1 = larger velocity in feet per second.
V_2 = smaller velocity in feet per second.

Where there is a sudden *decrease* in pipe diameter, the loss in head is given by

$$\Delta h = \frac{K\left(\frac{V_2}{V_1}\right)^2}{2g} \qquad (113)$$

where K is a constant factor depending on the ratio V_2/V_1 (Table XIV) and V_1 is again the larger of the two velocities.

TABLE XIV

VALUES OF K IN EQUATION (113)

$\frac{V_2}{V_1}$	K	$\frac{V_2}{V_1}$	K
0.1	0.469	0.6	0.257
0.2	0.431	0.7	0.212
0.3	0.387	0.8	0.161
0.4	0.343	0.9	0.079
0.5	0.298	1.0	0.000

Certain other losses are expressed as equivalent diameters of straight pipe, to be added to L in equation (110).

		Diameters
90° elbows	1 inch to 2½ inches	30
	3 " " 6 "	40
	7 " " 10 "	50
Globe valves	1 " " 2½ "	45
	3 " " 6 "	60
	7 " " 10 "	75
Gate valves		6 to 12
Tees, full sized branch		60
90° Curved pipe		
Center line radius = diam. of pipe		20
" " " = 2 to 8 diam.		10

In addition to all the above, the total power necessary to force a fluid through a pipe must also include the power necessary to create the velocity. This is called the *velocity head,* and may be calculated from the equation

$$\Delta h = \frac{V^2}{2g} \tag{114}$$

Appendix 2.

Test Code for Evaporating Apparatus.

FOREWORD.

Reorganization of the Power Test Codes Committee of the A S.M.E. in 1918 included the addition of a Committee to develop a Test Code for Evaporating Apparatus. In 1921 a Committee was appointed by the American Institute of Chemical Engineers which has closely coöperated with the A.S M.E. Committee in this work. The Code was printed in preliminary form in the March, 1921, issue of *Mechanical Engineering,* and was presented to the Society for discussion at a public hearing held in May, 1921, during the Spring Meeting in Chicago At the May, 1924, meeting of the Main Committee it was approved in its finally revised form and was later approved and adopted by the Council as a standard practice of the Society. It was approved and adopted by the American Institute of Chemical Engineers at their meeting in June, 1924.

Personnel of Power Test Codes Individual Committee No. 14 on Evaporating Apparatus.

E. N. Trump, *Chairman*

Burton N Bump, Consulting Engineering, 113 Circle Road, Syracuse, N Y
Ezra A. Newhall, Designing and Consulting Engineer, 1624 Pine Street, Philadelphia, Pa.
Harry L. Parr, Assistant Professor Mechanical Engineering, Columbia University, New York.
Lewis C Rogers, Power Engineer, The Solvay Process Co., Detroit, Mich
Edward N Trump, Vice-President, Solvay Process Company, Syracuse, N Y

Personnel of Committee of the American Institute of Chemical Engineers Coöperating with the A.S.M.E. Committee.

W L Badger, *Chairman*

W. L. Badger, Professor of Chemical Engineering, University of Michigan, Ann Arbor, Mich.
H. W. Dahlberg, Chief of Research, Great Western Sugar Company, Denver, Colo.
Crosby Field, Engineering Department, National Aniline and Chemical Company, Buffalo, N. Y.
Hugh K. Moore, The Brown Co., Berlin, New Hampshire.

TEST CODE FOR EVAPORATING APPARATUS.

Introduction.

1. This code for tests of evaporating apparatus is intended for use in testing single- or multiple-effect evaporators with or without vacuum in which water is boiled, driving vapors therefrom and separating it from the solution to be concentrated by the absorption of heat through metal walls from one fluid to another.

2. Steam and the vapor from the effects will usually be used as the heating medium.

Object.

3. The object of the test must be clearly outlined before the test is begun and the conditions held as uniform as possible to accomplish the purpose. If the object relates to the fulfilment of a contract-guarantee, an agreement should be made before the test between the interested parties concerning all matters about which dispute may arise, as noted in Par. 6 of the "Code on General Instructions," and the points agreed upon should be stated in the Report of the Test.

4. The purpose of the tests may be to determine one or more of the following:

(*a*) The adaptability of the apparatus to the solution to be concentrated and the best method of operation.

(*b*) The capacity or efficiency or both of a new installation preparatory to acceptance.

(*c*) To compare the efficiency of differences in design between the apparatus under test and some other for a similar purpose, and to establish relative constants for the type.

(*d*) To obtain data for cost accounting, production records and rate of heat transfer.

(*e*) To locate and eliminate losses by improving methods of operation or obtaining data leading to changes in design and construction.

Measurements.

5. The measurements involved in the testing of evaporating apparatus heated by steam will depend to some extent upon the object for which the test is conducted as defined in Par. 3 above. In general, these measurements will include the following:

(*a*) The general data relative to type of apparatus, and principal dimensions as defined in Table 1, Items 7 to 20.

(*b*) The weight or volume of the liquor before and after each effect, and for the apparatus as a whole.

(*c*) .The weight or volume of the steam to each effect and to the apparatus as a whole; also to any auxiliary apparatus.

(*d*) The pressure and temperature of the steam to each effect; also to any auxiliary apparatus.

(*e*) The temperature, specific heat, and specific gravity of the liquor before and after each effect, and before and after the unit as a whole.

(*f*) The weight or volume of water evaporated from the liquor.

(*g*) The weight or volume of condensing water used.

(*h*) A test for entrainment in all condensate.

Instruments and Apparatus.

6. The instruments and apparatus required for testing evaporating apparatus will include:

(*a*) Tanks and platform scales for weighing liquor and water, or calibrated tanks, or meters.
(*b*) Pressure and vacuum gages.
(*c*) Thermometers
(*d*) Calorimeters for determining the quality of steam.
(*e*) Hydrometers for determining the specific gravity of the liquors, and degree of concentration.
(*f*) Apparatus for analyzing the liquors.
(*g*) Barometer.
(*h*) Electrical instruments, including voltmeter, ammeter, and wattmeter, if electrical measurements are necessary.

Directions covering the application, use, calibration and accuracy of these instruments, and pieces of apparatus, are given in the various chapters of the "Code on Instruments and Apparatus."

Precautions.

7. A mercury-in-glass thermometer should always be corrected for exposed stem, especially when recording temperatures much above room temperature.

8. It is difficult to measure the temperature of superheated vapors with a mercury thermometer.

9. Mercury or oil wells for thermometers and an undue number of fins, nuts, bushings, etc., in the thermometer mountings reduce the thermometer readings by conducting away heat.

10. Spring gages are unreliable for pressures under 5 pounds gage. Pressures less than one atmosphere and up to 5 pounds gage must be measured with direct-reading manometers. Pressures from 5 pounds to 30 pounds should also be so measured wherever possible.

11. Venturi tubes, orifices, or other types of flow meters are not accurate for pulsating or intermittent flow such as, for instance, the discharge of a reciprocating pump. Such meters should therefore always be located after a large air chamber or to deliver with a constant pressure into an open tank from which the pumps take the feed to the apparatus.

Preparations.

12. The Power Test "Code on General Instructions" should be studied and followed as far as it relates to the particular test in hand The construction and operation of the apparatus also must be studied carefully to determine the many variables which may affect the results lest some apparently unimportant factor vitiate the test.

13. Establish definitely the importance of each variable quantity to the test and make careful arrangements for recording fluctuations in the readings of this quantity.

14. Determine and record whether or not the apparatus is to be tested "as found" or after being put in the best condition.

15. Note the details of construction, the arrangement of auxiliaries, and general course of boiling solution, condensates, and vapors with the apparatus in operation.

16. Record condition of heating surface, tightness, and freedom of moving parts.

Operating Conditions.

17. Determine the operating conditions necessary to conform to the object of the test, and make sure that these conditions are maintained throughout the test. (See Par. 23 of the "Code on General Instructions.")

Starting and Stopping.

18. The pressures, temperatures, densities and volumes of liquors within the evaporator should be the same at the beginning and at the end of the test. The test should be started at the beginning of a commercial cycle where such cycles exist in the operation of the evaporator, and after running the required number of hours it should be stopped at the beginning of the next succeeding cycle.

Duration.

19. The duration of the test should depend on the degree of accuracy demanded by the object of the test. If the apparatus regularly operates in cycles of less than four hours' duration at least three cycles should be included and, if one of these does not substantially corroborate the other two, additional tests should be run. If the operation operates continuously, the test should cover at least 12 hours of operation and should be prolonged if the conditions fluctuate.

20. The general rule is that the test must be so long that the inclusion or exclusion of a single extreme reading (due to inability to control conditions) on one quantity should not change the final average value of that quantity by an amount as great as the limit of accuracy expected for the test as a whole. In this connection recording thermometers and pressure gages are of great value even though the readings are not accurate enough to use for computation. They show whether or not conditions have been uniform and whether or not large deviations from the prescribed conditions are occurring between readings.

Records.

21. The data should be taken and recorded in the manner described in the General Instructions, Pars. 24 to 35. Quarter hourly readings of the various instruments will be sufficient, except where there is considerable fluctuation in such readings, in which case more frequent readings must be taken in order to give good averages. Uniformity

of operation may be assisted by the plotting of a chart while the test is in progress, as pointed out in Par. 33 of the "Code on General Instructions."

22. It is common experience that no set of test records, no matter how full, contains all the information that is later wanted. Every detail of operation, every deviation from established conditions, every unusual occurrence should be fully noted. So far as possible the original record should also be the permanent record, made in ink at the time of the reading

23. The data and results should be tabulated in accordance with the form shown in Table 1, forming part of this Code, adding items not provided for and omitting items not needed to conform to the object in view. Unless otherwise indicated, the items refer to the numerical averages of the readings which are recorded in the log.

Calculation of Results.

24. The methods of calculating the results as tabulated, so far as not self-evident, or not sufficiently indicated in the test for the items, are explained in the following section. The test of evaporating apparatus will ordinarily be concerned with the capacity of the apparatus, its efficiency, or both.

25. *Capacity.* The capacity of any equipment covered by this code is best expressed by its heat transfer coefficient, or the heat transfer coefficients of as many parts as can be tested separately.

26. Rating on any basis is affected by the temperature, absolute pressure, radiation losses, etc., maintained at the time the rating is being determined. Therefore, these conditions must be stated for every capacity determined.

27. If it is desired to rate the capacity of an evaporator on the basis of gallons evaporated per hour, this will only be accepted if stated in terms of standard United States gallons evaporated per hour from and at (temperature to be specified) with a total temperature drop of (to be specified) degrees.

28. *Capacity* (*General*). Heat transfer coefficients will be expressed as *apparent* coefficients. The apparent coefficient neglects the elevation in boiling point of the liquid evaporated. Its ratio to the true coefficient is the ratio of the true temperature drop to the apparent temperature drop. It may be computed by the formula:

$$U = \frac{H}{S \times q \times (t - t^1)} \qquad (1)$$

where U = apparent transfer coefficient in B t.u. per square foot per degree F. per hour.
H = total heat transmitted (in B.t.u.) in the time q.
S = heating surface in square feet.
q = time in hours.
$(t - t')$ = apparent temperature drop in degrees F.

29. *Capacity (Heating Surface).* Heating surface will be calculated as the surface in contact with the steam or other heating medium. In the case of multiple-effect evaporators, the coefficient will be calculated for each effect separately; hence heating surface as discussed here is heating surface of one effect.

30. *Capacity (Heat Transferred).* This will ordinarily be determined by computing the weight of steam used and the available heat per unit weight. A determination of the weight of the condensate is the most satisfactory method. This may be done by any convenient means, though direct weighing is most satisfactory. If this method is used, care must be taken to cool any condensate which may be above 212° F. to a point where no flash will occur during weighing. There are no meters which will handle hot water except the "flow meter" types, and these can only be used where the flow is steady and the pipe line in which they are used runs full. In those cases where non-condensed gases are handled through the condensate lines, direct weighing is the only method.

31. Condensate from effects other than the first must be tested for the presence of entrained liquid, and the corresponding corrections made.

32. It must be noted that the weight of the condensate from any effect is a measure of the heat transmitted in that effect, and not in the previous effect. The difference is due to self-evaporation in the case of forward feed, or to heating in case of backward feed. In exceptional cases, the weight of steam condensed in any effect may be determined from the difference in weight between liquor fed to the previous effect and liquor fed to the effect in question. This will seldom be as satisfactory as a direct determination of the weight of condensate.

33. *Available Heat.* This will ordinarily consist of:

(*a*) Superheat
(*b*) Latent Heat
(*c*) Sensible Heat.

34. Superheat will be difficult to measure, but is in all cases a small fraction of the total. If a thermometer in the steam inlet shows a temperature higher than that corresponding to the pressure, the heat as superheat should be calculated.

35. Latent heat should be taken for the temperature corresponding to the pressure of the steam rather than corresponding to its temperature reading, as the latter will usually be the less accurate reading. If the temperature of any part of the steam space is less than that corresponding to the pressure, air is not necessarily present since thermometers are very apt to read low due to radiation and conduction through their mountings. In any case, however, where the average temperature of the steam space is definitely shown to be lower than that corresponding to the average pressure, the presence of air or

other non-condensable gases is to be assumed and the latent heat taken as that corresponding to the temperature reading.

36. Sensible heat is determined from the difference between the temperature corresponding to the pressure in the steam space, and the temperature of the condensate leaving the steam space. Care must be taken to measure condensate temperature as near as possible to the last point at which it can give up useful heat.

37. In determining the total amount of heat transmitted, the *quality* of the steam used must be taken into account. In the case of steam going to the first effect, this may be done with any standard type of steam calorimeter, though it must be remembered that no type of steam calorimeter is reliable for small pressure drops. For steam passing from one effect to the other, practically the only moisture that can be determined is that present as entrainment, which will be found by an analysis of condensate.

38. *Apparent Temperature Drop.* The apparent temperature drop is the difference between the temperature of the heating steam calculated from its pressure, and the temperature of the vapors of the boiling liquid calculated from their pressure. This definition neglects the elevation of the boiling point of the liquid undergoing concentration and assumes the following conditions:

(*a*) that the steam space is sufficiently vented so the partial pressure of non-condensable gases is never high enough to have an appreciable effect on the steam temperature;
(*b*) that any elevation of steam temperature due to superheat corresponds to so little heat compared to the latent heat, that its effect on the mean temperature drop is negligible;
(*c*) that the effect of elevation of boiling point of the liquid being evaporated on the capacity of the apparatus is most conveniently expressed by using apparatus coefficients;
(*d*) that the effect of hydrostatic head is a characteristic of each machine, and is best expressed by apparent coefficients.

The use of pressures for this calculation is the principal reason for eliminating spring gages from such tests and relying only on direct-reading manometers.

As a matter of general information, the temperature of the boiling liquid is to be determined in each effect and made part of the records.

39. *Efficiency Defined.* A satisfactory method of expressing the efficiency of an evaporator is difficult to find. The true thermodynamic efficiency is zero, since an evaporator does no external work. Ratio of heat input to heat rejected is unsatisfactory because the heat going from the last effect to the condenser bears no direct or useful relation to the useful result accomplished, especially if vapors are withdrawn from earlier effects for other purposes. Probably the most useful, but not the simplest, rating is, ***pounds steam used per equivalent pound of water evaporated from and at*** (a specified temperature).

40. *Efficiency* (*Steam Used*). Steam supplied to the apparatus may be:

(*a*) exhaust from other parts of the plant.
(*b*) exhaust from auxiliaries of the apparatus being tested.
(*c*) live steam supplied direct to the apparatus.
(*d*) live steam supplied to auxiliaries of the apparatus being tested but not exhausted to it.

41. The first three items may be determined by a direct determination of the weight of condensate from the first effect, and the last item may be determined by weighing the condensate from the auxiliaries (liquor heaters, etc.). Live steam delivered to the apparatus and to its auxiliaries should be determined separately and reported as items distinct from the amounts of exhaust steam from other parts of the plant, since their heat content is different from the heat content of exhaust steam and the relative amounts of these two sets of items has a fundamental bearing on the actual cost of operation.

42. Steam supplied as live steam to an evaporator, steam-turbine-driven auxiliary, heater, or other device taking steam at a uniform rate may be determined by any form of "flow meter" based on a Venturi tube, orifice, or other primary device. Live steam supplied to reciprocating pumps or reciprocating engines driving auxiliaries cannot be accurately determined by flow meters, but must be measured by special tests of the unit in question.

43. Steam charged to the evaporator will, in general, consist of all live steam supplied to the evaporator and its auxiliaries (whether these auxiliaries condense it or not, provided that if they do not condense it, it is returned as exhaust to the evaporator), plus whatever exhaust is supplied from sources other than its own auxiliaries. The difference between the heat units entering and leaving the evaporator in either steam or hot water should be charged to it. In case an evaporator furnishes steam at any pressure to other parts of the process (such as vapors for heating in other parts of the plant not directly concerned with evaporation, or vapors for operating other evaporation units not involved in this test), the question of whether or not this steam is to be credited to the evaporator, and on what basis, will depend on the method of accounting used in that particular process. If desired, the evaporator may be credited with the amount of live steam that would be necessary to do the same work that is here performed by vapors. No general rule can be established here. Each case must be decided separately.

44. *Efficiency* (*Total Evaporation*). The total weight of water evaporated by the whole evaporator or by any one effect may be determined by weighing the total liquid entering and leaving. For cases where this is inconvenient, the liquid entering and leaving may be measured with a flow meter, provided a representative sample is taken and analyzed for total solids. It must be noted that determining total solids from density readings is not satisfactory in many cases, and the

"density-solids" curve is not known at all for many liquids handled commercially.

45. A much more accurate determination will generally be obtained by measuring or weighing the condensate from the second and all succeeding effects, and adding to the sum of such weights the weight of vapor going from the last effect to the condenser. In these weights must be included the weight of condensate from any apparatus heated by vapor from the evaporator body in question. Vapor to the condenser may be determined by an orifice in the vapor line, or by measuring the cold water fed to the condenser and determining its rise in temperature, or by weighing or measuring the feed into and out of the last effect If the second method is used, the weight of vapor is to be determined by

$$V = \frac{W\,(t_2 - t_1)}{L - (t_3 - t_2)} \qquad (2)$$

where V = weight of vapor going to the condenser.
W = weight of cold water fed to the condenser.
t_1 = the temperature of cold water going to the condenser.
t_2 = temperature of the hot water leaving the condenser.
L = latent heat of steam at the pressure of the vapor entering the condenser.
t_3 = temperature of the vapor going to the condenser.

In all calculations of weight of water evaporated, condensates should be tested for entrainment and the proper corrections made

46. *Efficiency (Total Equivalent Evaporation)* Total evaporation as determined in the paragraph above is not a measure of the work done in an evaporator, for two evaporators might do the same *total* evaporation, but owing to differences in feed temperature might require different amounts of steam. Also, if one evaporator works at a high boiling point and another at a low boiling point, changes in the latent heat of evaporation will alter the relations between the work accomplished in the two cases and the steam necessary.

47. The real work done by an evaporator other than removal of water is to heat the thick liquor or discharge from the temperature at which thin liquor is fed to the temperature at which thick liquor leaves the evaporator. If the feed temperature is higher than the exit temperature, this quantity will be negative and is to be so used in the calculations.

48 Since within the range ordinarily covered by evaporators, the latent heat varies but slightly, the total evaporation as determined above will change but little when converted to evaporation from and at a specified temperature. If this result is desired, it is to be obtained by the following formula:

$$\text{Equivalent evaporation} = \frac{W_1 L_1}{L} + \frac{W_2 L_2}{L} + \dots + \frac{W_t\,(t_3 - t_4)}{L} \qquad (3)$$

where W_1, W_2, etc. = weights of water actually evaporated in the 1st, 2nd, etc., effects.

W_t = weight of thick liquor leaving the evaporator (or leaving any heat interchanger, etc., which returns the heat it recovers to the evaporator, in some form).

L_1, L_2, etc. = latent heats of evaporation at the pressure existing in the vapor spaces of the 1st, 2nd, etc., effects.

L = latent heat of evaporation at the temperature taken for reference.

t_3 = temperature of the thick liquor leaving the system.

t_4 = temperature of the feed entering the system.

In this connection it must be noted that any heaters or heat interchangers adding heat to the feed or the thick liquor or taking heat from the feed or thick liquor are part of the evaporator system.

If it is not considered necessary to convert the evaporation in the different effects to a common temperature, the quantity L in the last term of equation (3) is to be the mean latent heat throughout the evaporator system.

49. *Heat Balance.* An overall heat balance of a multiple effect evaporator is not easy to establish and is of doubtful value. A heat balance across each body and its accessories is very instructive. It consists, essentially, of equating heat entering to heat leaving plus radiation and other unknown losses.

50. *Heat Entering.* The heat entering each effect is made up of—

(*a*) Heat entering in steam (or vapors).
(*b*) Heat entering in liquid.

The heat entering in steam is the weight of the condensate from the effect in question multiplied by the total heat of steam at the pressure existing in the steam chest or calandria. The heat entering with the liquid is the weight of liquid entering multiplied by its specific heat and its temperature above the datum on which the heat balance is based. The weight of liquid entering any effect is best determined by subtracting from the total weight fed to the first effect the weights of condensate from the effect in question and all preceding effects, except the first, but including the condensate from all heaters, etc., heated by vapors from the effects included. Where "Density-Solids" curves are available for the liquor in question, weight of water evaporated may be determined by flow meters between the effects, but this method is inferior to the preceding.

51. *Heat Leaving.* The heat leaving each effect may consist of—

(*a*) Heat leaving in liquor to the next effect.
(*b*) Heat withdrawn in hot products (such as crystals removed, etc.).

(*c*) Heat leaving in vapor caused by regular evaporation.
(*d*) Heat leaving in vapor caused by self-evaporation.
(*e*) Heat leaving in condensate, and
(*f*) Heat lost by radiation

52. Heat leaving in liquor to next effect is to be determined in the same manner as heat entering in the liquor, as described in the preceding section. Heat withdrawn in hot product is determined by determining the weight of such material withdrawn, its temperature above the datum, and its specific heat. Heat leaving in condensate is determined from weight and temperature of condensate. Items (*c*) and (*d*) above cannot be separated, and in fact do not need to be. Their sum is the weight of condensate in the next effect (plus condensate from any heaters operated by vapors from the effect in question) multiplied by the total heat of steam at the pressure of the vapor space of the effect in question Radiation and other losses are then the difference between total heat entering the effect and total heat accounted for leaving the effect.

53. Radiation for the whole evaporator is determined by repeating such a heat balance as the above for all effects and all auxiliaries.

54. *Thermocompression Evaporators.* Such evaporators are uncommon, but it should be pointed out that while determinations of capacity may be made exactly as above, efficiency cannot be determined as above, for all the steam entering the first effect is *not* steam supplied to the system.

Steam used will consist of—

(*a*) Exhaust or live steam supplied from external sources, and
(*b*) The steam equivalent to the power used in compression.

If the compressor is driven by a steam engine or turbine, the steam used by such engine or turbine is to be charged to the evaporator unless exhaust is sent from them to other places than the evaporator system. In such a case the evaporator is to be credited with the steam so withdrawn. If the compressor is motor or belt driven, the power consumed is to be determined and the equivalent of this power expressed as steam is to be charged to the evaporator.

55. *Corrected Coefficients.* It is desirable to report not only apparent coefficients as defined in Par. 28, but coefficients corrected in various ways.

56. *Hydrostatic Head Correction.* The static level of liquid in the evaporator should be determined and recorded regularly, and the record should be referred to such a datum that the mean hydrostatic head of liquor on the heating surface may be calculated. By adding this mean hydrostatic head to the pressure in the vapor space, the mean total pressure on the heating surface is obtained, and from this the mean temperature of the liquid in contact with the heating surface. If this temperature be used to determine t_1 in Equation (1), (Par. 28) instead

of the temperature calculated from the pressure of the vapor space, the resulting coefficients are said to be *corrected for hydrostatic head.*

57. *Elevation of Boiling Point Correction.* In general the boiling point of most liquids handled in practice will be higher than that of pure water at the same pressure. This causes a decrease in capacity of the evaporator which is usually represented by using *apparent* coefficients. For studying the effect of variations in design, it is desirable to correct apparent coefficients for this elevation. If actual boiling temperatures, as taken with a thermometer, are used for calculating t_1 in Equation (1), Par. 28, instead of temperatures corresponding to pressures in the vapor space, the resulting coefficients are said to be *corrected for elevation of boiling point.* In making such a determination of temperature, care must be taken that the thermometer is not affected by currents of liquor entering the body at temperatures other than that of the boiling liquid, or by hydrostatic head.

Detailed Directions.

58. *Excess Boiling Degrees.* (Table 1, Item 34.) The excess boiling degrees of the liquor in any effect is the difference between the boiling temperature of the liquor in that effect and the boiling temperature of water under the same pressure as that of the vapor in the effect.

59. *Steam to Condenser.* (Table 1, Item 52.) This cannot be measured directly. If a surface condenser is used this item can be determined by measuring or weighing the condensed steam discharged from the condenser. If a jet condenser is used, Item 52 may be calculated from the weight of condensing water and the rise in temperature of the condensing water. The product of these two will give the heat transferred to the condensing water, and this heat divided by the heat in a pound of steam going to the condenser, reckoned from the temperature at which the condensing water leaves the condenser, will give the weight of steam going to condenser. The same item may be determined by taking the difference in weight between feed and discharge liquors in the last effect. If any retained product is found in the steam to the condenser, this must be taken account of in calculating the weight of steam going to condenser.

60. *Evaporation from Liquor.* (Table 1, Item 53.) This is the sum of the steam to the condenser (Item 52) and the condensed steam drained from the effects, except that from the first effect, plus any vapor drawn off for heaters or other purposes.

61. *Liquor Entering.* (Table 1, Item 56.) The weight of the liquor entering first effect is the weight of the feed liquor.

The weight of the liquor entering the following effects will be calculated by subtracting the condensate in said effects plus all previous effects except the first, including condensate bled to any outside heater In case it is possible to weigh or measure accurately the liquor passing into each effect, these calculations may be thus checked.

62. *Evaporation from Liquor in Individual Effects* (Table 1, Item 58) The satisfactory determination of evaporation in the several effects depends on the successful weighing or measuring of the condensate or the liquor to each effect (Items 56 and 57), corrected for self-evaporation or entrainment.

63 *Evaporation from Liquor from and at 212° F.* (Table 1, Item 54) This is Item 68 ÷ 970.4.

64 *Self-Evaporation* (Table 1, Item 59) Self-evaporation in any effect is determined as follows Multiply the weight of the liquor entering that effect by the temperature drop between the boiling point in that effect and the boiling point in the preceding effect, and by the specific heat of the liquor. Divide this quantity by the latent heat of evaporation in the effect considered The result is the weight of self-evaporation.

65 *Heat Contained in Entering Liquor* (Table 1, Item 63.) This is the pounds of feed liquor multiplied by the temperature above 32° F. of the feed liquor, and this product multiplied by the specific heat of the feed liquor.

66 *Heat from Steam Supplied to Entire Apparatus.* (Table 1, Item 64.) This is the sum of the products of the pounds of steam, from Items 42, 43 and 44, multiplied respectively by the total heat above 32° F., in a pound of steam of the pressures and qualities used

67. *Heat to Pumps.* (Table 1, Item 65.) This is the product of the pounds of steam to pumps (Item 44), by (the heat units in a pound of steam to pumps minus the heat units in one pound of exhaust steam from pumps corrected for quality).

68. *Heat Contained in Discharge Liquor.* (Table 1, Item 66.) This is the product of the pounds of discharge liquor (Item 51) by the temperature, above 32° F., of the discharge liquor, and this product multiplied by the specific heat of the discharge liquor.

69. *Heat in Steam to Condenser.* (Table 1, Item 67.) If the quality of this steam can be obtained this item may be calculated from the weight of steam to condenser (Item 52) and the heat units in a pound of this steam; otherwise this item must be calculated from the heat absorbed by the condensing water; or, pounds of condensing water (Item 62) multiplied by rise in temperature of condensing water [Item 37 (*b*)—37 (*a*)]. To this must be added the heat in the condensate above 32° F.

70 *Heat Contained in the Evaporation from Liquor.* (Table 1, Item 68.) This is the heat in the steam going to the condenser plus the heat in the condensed steam drained from the effects (except that drained from the first effect: Item 67 + (Item 73 — Heat in condensed steam from first effect).

71. *Heat Contained in Entrained Product.* (Table 1, Item 69) From the analysis of the steam to the condenser (Item 92) the amount of entrained product is determined. The weight of this entrained product, multiplied by its temperature above 32° F. and this product,

multiplied by the specific heat of the entrained product, will give the heat in such product escaping with steam to condenser.

72. *Heat Supplied to Heaters.* (Table 1, Item 70) This is the product of the pounds of steam to heaters (Item 42) by the total heat above 32° F. in a pound of such steam. In determining the heat in the steam the quality of the steam must be taken into account.

73. *Heat in Water from Heaters.* (Table 1, Item 71.) This is the product of the pounds of water discharged from heaters (Item 60) by the temperature of such water above 32° F.

In case of more than one heater the heat will need to be calculated for each heater separately and the sum of the several quantities taken.

74. *Heat Supplied to the Effects.* (Table 1, Item 72.) This is the product of the pounds of live steam to the effects (Item 43) by the total heat above 32° F. in a pound of such live steam. If exhaust steam is supplied to the effects its heat must be added to the above (pounds of exhaust steam (Item 48) multiplied by the total heat, above 32° F. in a pound of exhaust steam).

75. *Heat in Condensed Steam Drained from Effects.* (Table 1, Item 73.) This is condensate taken from the steam space of an effect. In case of more than one effect this may be passed along to the next effect, where its heat may, in part at least, be transferred to the liquor. This heat is the product of pounds of condensed steam [Item 57 (*b*)] multiplied by the temperature above 32° F. of such condensate. In case the apparatus has more than one effect the heat must be calculated for each effect separately, and the sum of the several quantities taken.

In case the condensed steam flows from the steam space of one effect to that of the next effect its temperature should be measured at point of leaving the last effect of the series. This temperature, in degrees above 32° F., multiplied by the pounds of condensed steam leaving the last effect will give the heat in the total condensed steam drained from the effects.

76. *Heat Discharged from the Effects.* (Table 1, Item 74.) This is the sum of Items 66, 67, 68 and 73.

A.S.M.E. POWER TEST CODES

TEST CODE FOR EVAPORATING APPARATUS

TABLE 1

DATA AND RESULTS OF EVAPORATING APPARATUS TEST

GENERAL INFORMATION

1	Date of test	
2	Location	
3	Owner	
4	Builder of apparatus	
5	Test conducted by	
6	Object of Test	

DESCRIPTIONS, DIMENSIONS, ETC.

7	Kind and type of apparatus under test (vacuum pan, multiple effect evaporator, etc.)	
8	Dimensions of shells	
9	Material of tubes and tube sheets	
10	Number of tubes in each effect—Length	
	Diameter	
11	Heating surface in each effect	sq. ft.
12	Number and type of liquor heaters	
13	Number of tubes in each heater—Length	
	Diameter	
14	Heating surface in each heater	sq. ft.
15	Liquor-heating surface in each heater	sq. ft.
16	Type and make of condenser equipment	
17	Condensing surface	sq ft.
18	Type and dimensions of vacuum and circulating pumps	
19	Kind of liquor concentrated	
20	Characteristics of liquor concentrated	

TEST DATA AND RESULTS

			Test No. 1	Test No. 2
21	Duration of test	hr.		
	Average Pressures.			
22	Steam pressure by gage	lb. per sq. in.		
23	Barometric pressure	in. of mercury		
24	Absolute steam pressure	lb. per sq. in.		
25	Pressure in exhaust-steam line from pumps	lb. per sq. in.		
26	Pressure or vacuum in each effect in lb. pressure and inches of vacuum			
27	Vacuum in the condenser	in. of mercury		
	(*a*) Corresponding absolute pressure	lb. per sq. in.		
28	Liquor level in each effect	inches		
	Average Temperatures.			
29	Temperatures of saturated steam corresponding to the vapor pressure in each effect	° F.		
30	Steam temperatures:			
	(*a*) Superheated steam	° F.		
	(*b*) Normal temperature of saturated steam	° F.		
	(*c*) Temperature of exhaust steam	° F.		
31	Temperature of liquor entering each heater	° F.		
32	Temperature of liquor entering each effect	° F.		
33	Temperature of liquor leaving each effect	° F.		
34	Excess boiling degrees of liquor in each effect	° F.		
35	Temperatures of condensed steam discharged from each heater	° F.		
36	Temperatures of condensed steam drained from each effect	° F.		
37	Temperature of condensing water:			
	(*a*) Entering condenser	° F.		
	(*b*) Leaving condenser	° F.		

TEST DATA AND RESULTS—(*Continued*)

38	Temperature of air in evaporating room	° F.
39	State of weather:	
	(*a*) Temperature of external air	° F.
	(*b*) Relative humidity of external air	per cent

Quality of Steam.

40	Live Steam:	
	(*a*) Moisture in steam	per cent
	(*b*) Superheat	° F.
	(*c*) Factor of correction for quality of steam	
41	Exhaust Steam:	
	(*a*) Moisture in steam	per cent
	(*b*) Factor of correction for quality of steam	

Total Quantities.

		Total Pounds	Pounds Per Hour	Test No. 1	Test No. 2
42	Steam to heaters				
43	Steam to effects				
44	Steam to pumps				
45	Total steam supplied to entire apparatus (sum of Items 42, 43 and 44)				
46	Equivalent weight of total steam from and at 212° F.				
47	Exhaust steam to heaters				
48	Exhaust steam to effects				
49	Total exhaust steam				
50	Feed liquor				
51	Discharge liquor or concentrated product				

52	Steam to condenser	
53	Evaporation from liquor	
54	Equivalent evaporation from liquor from and at 212° F.	
55	Equivalent evaporation from liquor, from and at 212° F. per sq. ft. of the total heating surface in all the effects	
56	Liquor entering each effect	
57	Liquor leaving each effect	
58	Evaporation from liquor in each effect	
59	Self-evaporation in each effect	
60	Water discharged from each heater	
61	Condensed steam drained from each effect	
62	Condensing water	
63	Heat above 32° F. contained in entering liquor	B.t.u.
64	Heat from steam supplied to entire apparatus, including pumps. (Calculated from total live steam)	B.t.u.
65	Heat to pumps	B.t.u.
66	Heat above 32° F. contained in discharge liquor	B.t.u.
67	Heat contained in steam to condenser	B.t.u.
68	Heat contained in the evaporator from liquor	B.t.u.
69	Heat contained in entrained product escaping with steam to condensers	B.t.u.
70	Heat supplied to the heaters	B.t.u.
71	Heat above 32° F. in water discharged from heaters	B.t.u.
72	Heat supplied to effects by live and exhaust steam	B.t.u.
73	Heat above 32° F. contained in the condensed steam drained from the effects	B.t.u.
74	Heat discharged from the effects	B.t.u.

Evaporation Ratio and Efficiency of Individual Effects.

75	Overall thermal efficiency of entire equipment	per cent
76	Thermal efficiency of each effect	per cent
77	Thermal efficiency of the heaters	per cent

Capacity.

78	Evaporation per hour from and at 212° F.	lb.
79	Raw liquor to apparatus per 24 hours	gal.
80	Concentrated product per 24 hours	gal.
81	Apparent heat-transmission coefficients for each effect	
82	Corrected heat-transmission coefficients for each effect	

Economy.

83	Evaporation per pound of total steam used	lb.
84	Concentration product per pound of total steam used	lb.
85	Evapporation per pound of steam from and at 212° F.	lb.

Heat Balance.	Test No. 1 B.t.u.		Test No. 2 B.t.u.	
	per hour	per cent	per hour	per cent
86 Heat to pumps, heaters and effects:				
(*a*) Heat from steam to entire equipment (Item 64)				
(*b*) Heat above 32° F. contained in entering liquor (Item 63)				
Total = (*a*) + (*b*)				
87 Heat discharged from heaters and effects:				
(*a*) Heat in water from heaters (Item 71)				
(*b*) Heat in condensed steam drained from effects (Item 73)				
(*c*) Heat in steam to condenser (Item 67)				
(*d*) Heat in entrained product (Item 69)				
(*e*) Heat in discharge liquor (Item 66)				
Total = (*a*) + (*b*) + (*c*) + (*d*) + (*e*)				
88 Radiation and unaccounted for losses: [Item 86 (Total) — Item 87 (Total)]				

Liquor Analysis.

89 Analysis of feed liquor
90 Analysis of discharge liquor
91 Analysis of liquor out of each effect
92 Analysis of steam to the condenser to determine the amount of entrained product

Physical Properties of Liquor.

93 Specific gravity of feed liquor
94 Specific gravity of discharge liquor
95 Specific gravity of liquor out of each effect
96 Specific heat of feed liquor
97 Specific heat of discharge liquor
98 Specific heat of liquor out of each effect

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